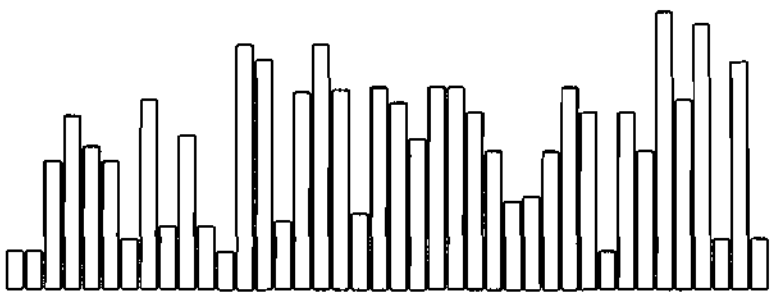
Certificate Course
on
PIPING ENGINEERING

June 12 - 25, 2006



Organized by

Piping Engineering Cell
Computer Aided Design Centre
Indian Institute of Technology, Bombay
Powai, Mumbai - 400 076

40th

Certificate Course on Piping Engineering

June 12-25, 2006

Piping Engineering Cell, CAD Centre, IIT Bombay

At

Dr. Babasaheb Ambedkar Technological University, Lonere

Monday, June 12, 2006

1030:1100	Inauguration
1100:1130	Tea
1130:1330	Introduction to Piping Engineering
1300:1400	Lunch
1400:1600	Introduction to Piping Engineering
1600:1630	Tea
1630:1730	Introduction to Piping Engineering

Tuesday, June 13 to Saturday, June 24, 2006

There will be four sessions every day (except on June 21 which is a rest day in the course).

0930:1100, 1130:1330, 1430:1600, 1630:1730

The tentative topics on these days are as follows.

Tuesday, June 13, 2006	Pipe Sizing
Wednesday, June 14, 2006	Mechanical Design Fundamentals
Thursday, June 15, 2006	Tutorials and Practical on Pipe Design
Friday, June 16, 2006	Codes & Standards, Piping Elements
Saturday, June 17, 2006	Valves, Basics of Drawing
Sunday, June 18, 2006	Plot Plan, Equipment & Piping Layout
Monday, June 19, 2006	Transient Flow, Pipe Under Stress, Tutorials
Tuesday, June 20, 2006	Nozzle Reinforcement, Cross-country Pipeline
Wednesday, June 21, 2006	Excursion/Rest Day
Thursday, June 22, 2006	Flexibility Analysis
Friday, June 23, 2006	Flexibility Analysis
Saturday, June 24, 2006	Support Selection & Design, Expansion Joints, Jacketed Piping Design

Sunday, June 25, 2006

0930:1130	Quiz and Feedback
1200:1300	Closing Remarks
1300:1330	Certificate Distribution

INTRODUCTION

About two decades ago, in India, the design procedure for piping systems for Refineries, Petrochemicals and Fertilizer Plants, in magnitude, depth and complexities were not fully evolved. Only in the recent past, we were exposed in detail to this field. Now we are self-sufficient in the field of piping technology and design.

Piping systems in a chemical plant are comparable to the vanes and arteries through which fluids, vapors, slurries, solids, etc. flow under various conditions, as imposed by the process design of the plant. Piping network is subjected to almost all the severest conditions of the plant such as high temperature, pressure, flow and combination of these. In addition to the above, corrosion, erosion, toxic conditions and radioactivity add to more problems and difficulties in piping design. With the process conditions becoming more and more severe by the advancement in process development, a continuous effort is required to be carried on simultaneously to cope up with the demands of process. This makes the job of a piping engineer more complex and responsible.

Piping, because of its nature, requires a number of day to day decisions on matters of detail, which, in some ways are often more difficult to solve than major issues connected with the project. It is this same detail which can cause expensive delays in design and construction and consequently in commissioning. All too often in the past, piping has been regarded as an unimportant job in the overall project engineering instead of being treated as a function requiring as wide a knowledge, experience and variety skills as any other branch of engineering.

In almost all chemical industries, the installed capital cost of piping is a major factor in plant investment. Figs. 1 and 2 show a chart based on oil refineries, chemical and petrochemical complexes. Fig. 1 shows that, excluding major equipment costs, piping is the largest plant cost component. It exceeds the next largest component by a factor of two. It is also observed from Fig. 3 that piping exceeds all other field costs by a substantial amount. Fig. 2 indicates that design engineering utilizes approximately 45% of engineering man-hours and 50% of these hours are used in piping design.

In addition to the above, the lost time in piping has an effect, which goes well beyond its direct cost, as it involves financial loss in some proportion to the total plant investment. The delay in and during start up means idle capital and losses in plant earning capacity.

In the recent years, the trend is to develop better techniques so as to save time in piping activities. Computer is being used extensively to obtain rapid solutions to the more complex problems of plant design and, in so for as piping is concerned, to the solving of problems of pipe stressing. More recently, it is being employed for production of piping detail drawings, piping isometrics, bill of materials, cost estimation and control. Piping engineer has therefore a further responsibility in understanding and application of continually growing techniques of this nature.

WHAT A PIPING ENGINEER SHOULD KNOW ABOUT

Piping engineer requires not only wide engineering knowledge - not necessarily in depth, but certainly in understanding - but he must also have an understanding of engineering economics and costs, of metallurgy, of methods of pipe fabrication and erection. He must have some knowledge of industrial chemistry and chemical engineering in addition to a sufficient knowledge of mechanical, civil, electrical and instrument engineering so as to discuss requirements and problems with specialists in these fields. This will be more clear with the data piping department requires from other disciplines as given in Annexure A. He should be co-operative, able to communicate effectively, lead or take part in teamwork, be alive to the application of new methods, materials and designs. He must be aware of standards, codes and practices.

There are several aspects of engineering technology, which the piping engineer must know something about - at least sufficiently to discuss rationally, any particular subject with specialists concerned. More importantly, he must have sufficient broad knowledge to know that certain conditions can arise at the early stages of plant design, where lack of awareness can cause difficulties and even disasters.

A fairly good knowledge of structural engineering is a must. Piping in operation is always in movement and subjected to pressure and forces with consequent reactions on mechanisms such as pumps, compressors and equipment in general, and on structures and related piping. Lack of knowledge can cause errors sufficient to cause machine or equipment breakdown or to overstress and even cause collapse of structures.

A good knowledge of safety codes and practices is also essential.

Above all, a piping engineer should be very well conversant with drafting procedures and practices.

PIPING DESIGN FUNCTION IN A SMALL AND A LARGE ORGANIZATION

The size and scale of a company or design office do not change either the basic piping design requirements or the underlying design principles and practices. As the volume of piping design work and the number of projects executed increases, so also does the degree of specialization increase. Functions such as piping layout, piping specifications, detail drafting and material listing begin to emerge as separate departments within the design office. There is little apparent similarity between piping design organization in a small office of say, ten men, and a large company with hundreds of piping men, as indicated in Figs. 4 and 5, but the job performed in both cases is identical and has the same degree of relative importance to the whole project design organization. The real difference lies in rate of projects passing through the design office. In the small office, the flow of projects is such that each project is substantially completed before the succeeding project commences. This demands flexible organization of a

small group, who have responsibility for all aspects of design. In the large company, however, projects are much more frequent. Thus the greater volume of work handled by a large group makes a high degree of specialization necessary for economic and administrative reasons.

In both cases, piping design is the most lengthy and complex part of the whole design procedure and almost always on the critical path of the project plan.

PIPING WITHIN THE PROJECT PLAN

Piping is an important element of every stage of project design, purchasing and construction. It is intimately linked to the other project work on equipment, electrical, instrument and civil engineering. Work on piping is proceeding at every stage of the project, partly because of the sheer volume of design and erection work, but mainly because of the need to relate other project activities to the piping design. A typical activity network and bar chart for a process plant project are shown in Figs. 6 and 7.

The activity network shows clearly the interrelated activities of piping and other branches of project, particularly at design stage. It illustrates the unifying role played by the piping engineer in acting as a clearinghouse for data provided from other specialist engineers. It also indicates the last stage of the project, when piping erection is being carried out and when one faces the difficulties of correcting the design errors without delaying project completion. Since the quantum of work involved in piping is very extensive, the design and erection activities of piping appear on the critical path of this network. The critical path planning helps in proper allocation and utilization of manpower for critical activities, thus avoiding uneconomic and wasteful allocation of resources.

RESPONSIBILITY OF A PIPING ENGINEER

"The efficiency with which the entire systems of any particular projects work depends upon initial phase and is in the hands of piping analytical engineer. Every change the engineer makes from his base design is compounded ten folds down stream as so many other operations depend on his design. The analytical design is the 'BIBLE' and must be correct in the first instance. The correctness, thoroughness and efficiency of the design released by the analytical engineer determines the efficiency of the piping design and influences, significantly, the efficiency and quality of overall plant."

Piping engineer is responsible for a substantial part of total project cost. Achievement of his program timings is critical to the completion of project in time. In addition to his own function as a specialist design engineer, he must provide a considerable amount of information and design continuity inside the project design organization.

HIS PARTICULAR CONCERNS MUST BE

Adequacy

- Piping design must be adequate to meet the process specification and physical conditions in which the plant is to operate.

Economy

- Adequate design must be achieved at an economic cost within the project budget. Design cost must be minimized by maximizing the use of standardized methods of detailing and data presentation.

Clarity

- Much of the piping data is derived from and used by other engineering departments and must be clear, consistent and reliable.

Accuracy

- Details of piping and materials must be complete and accurate. Rectification of mistakes in these activities at a later stage may prove to be very costly and can delay project completion.

The piping engineer has, therefore, considerable responsibility for economic and accurate design. Much development of design methods and organization has taken place over the years. In large design organizations, techniques have emerged, whereby procedures for producing simple symbolic data conveying maximum information at minimum cost can be employed. The basic elements of these procedures are:

- Rapid data retrieval
- Standardization of engineering design methods for stressing, material selection etc.
- Maximum use of standards
- Symbolic drawing procedures
- Standardization of document format for issue of piping information.

A responsible piping engineer would try to take the maximum use of such methods and procedures.

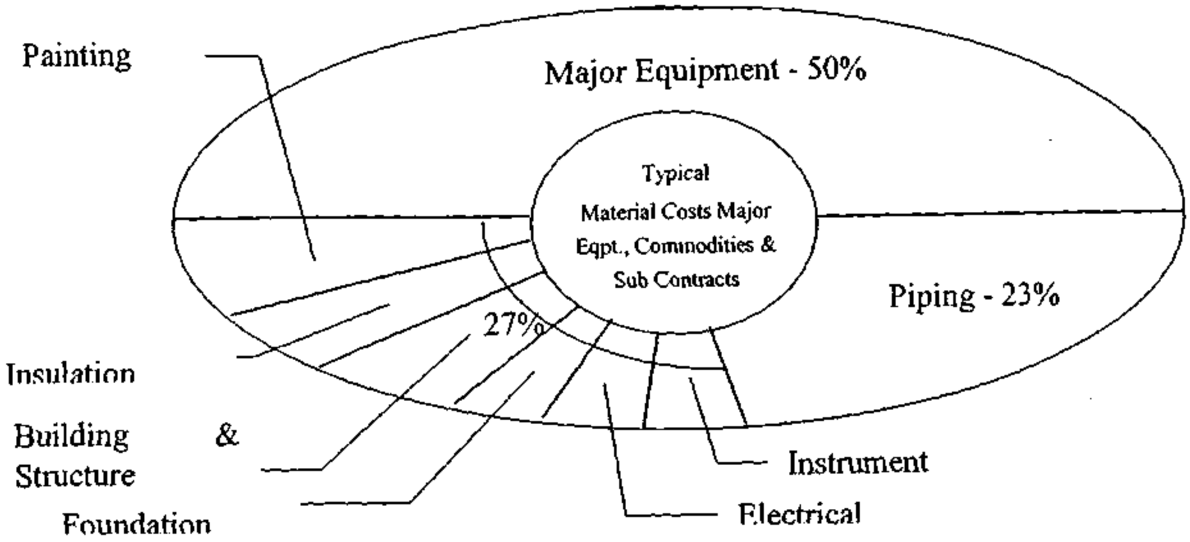


FIG. 1

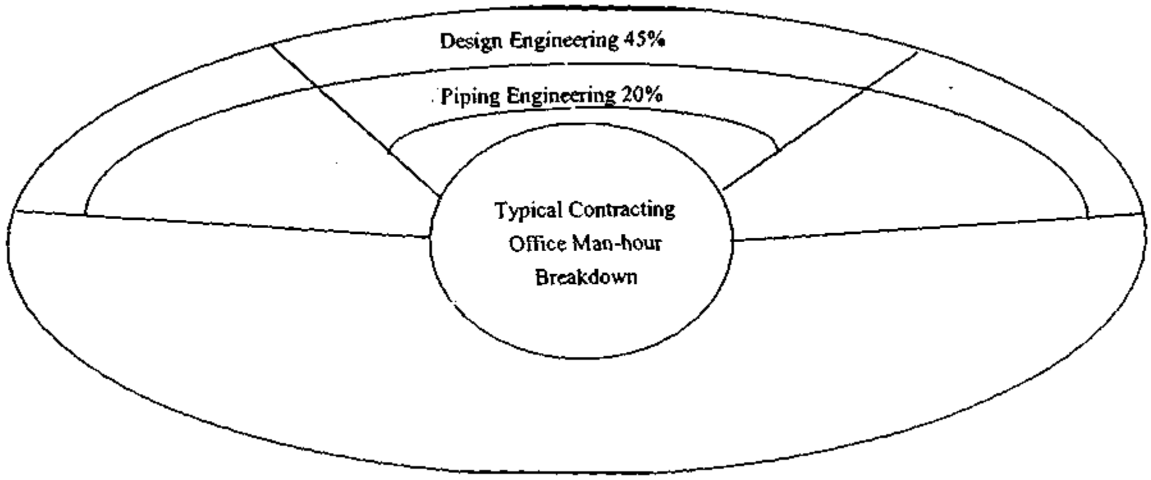


FIG. 2

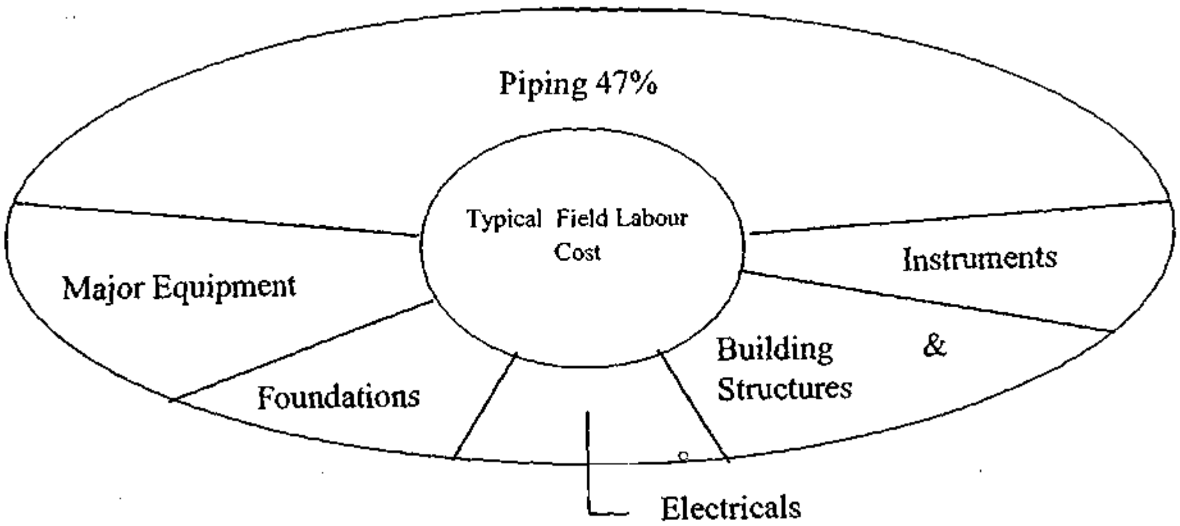


FIG. 3

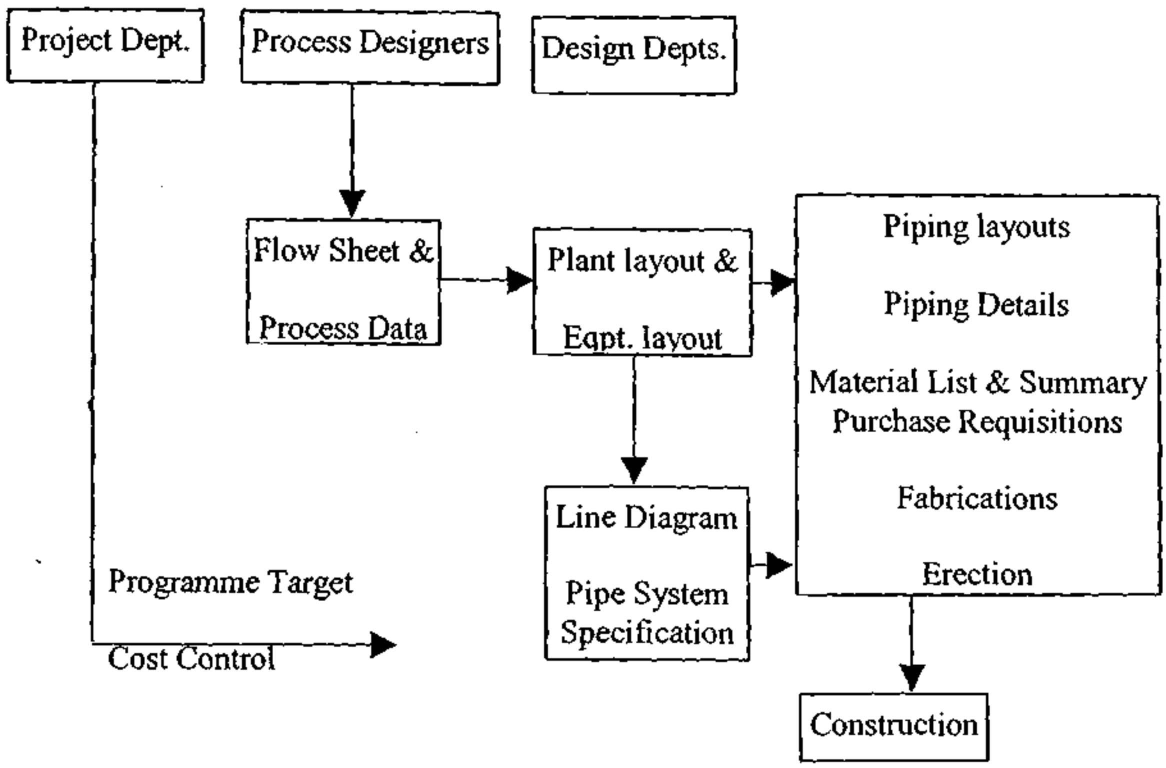


FIG. 4 PIPING DESIGN IN SMALL ORGANIZATION

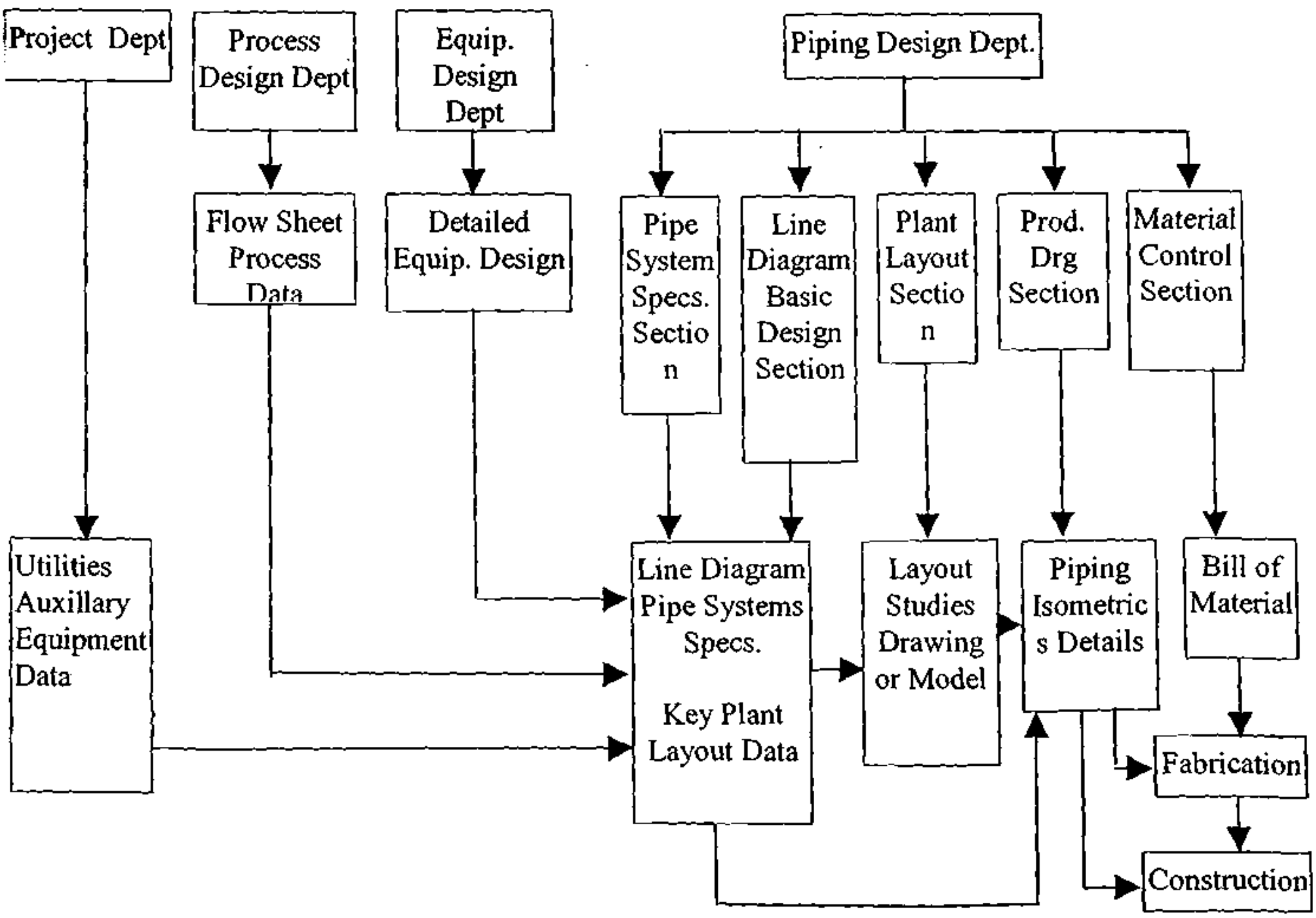


FIG. 5 PIPING DESIGN IN LARGE ORGANIZATION

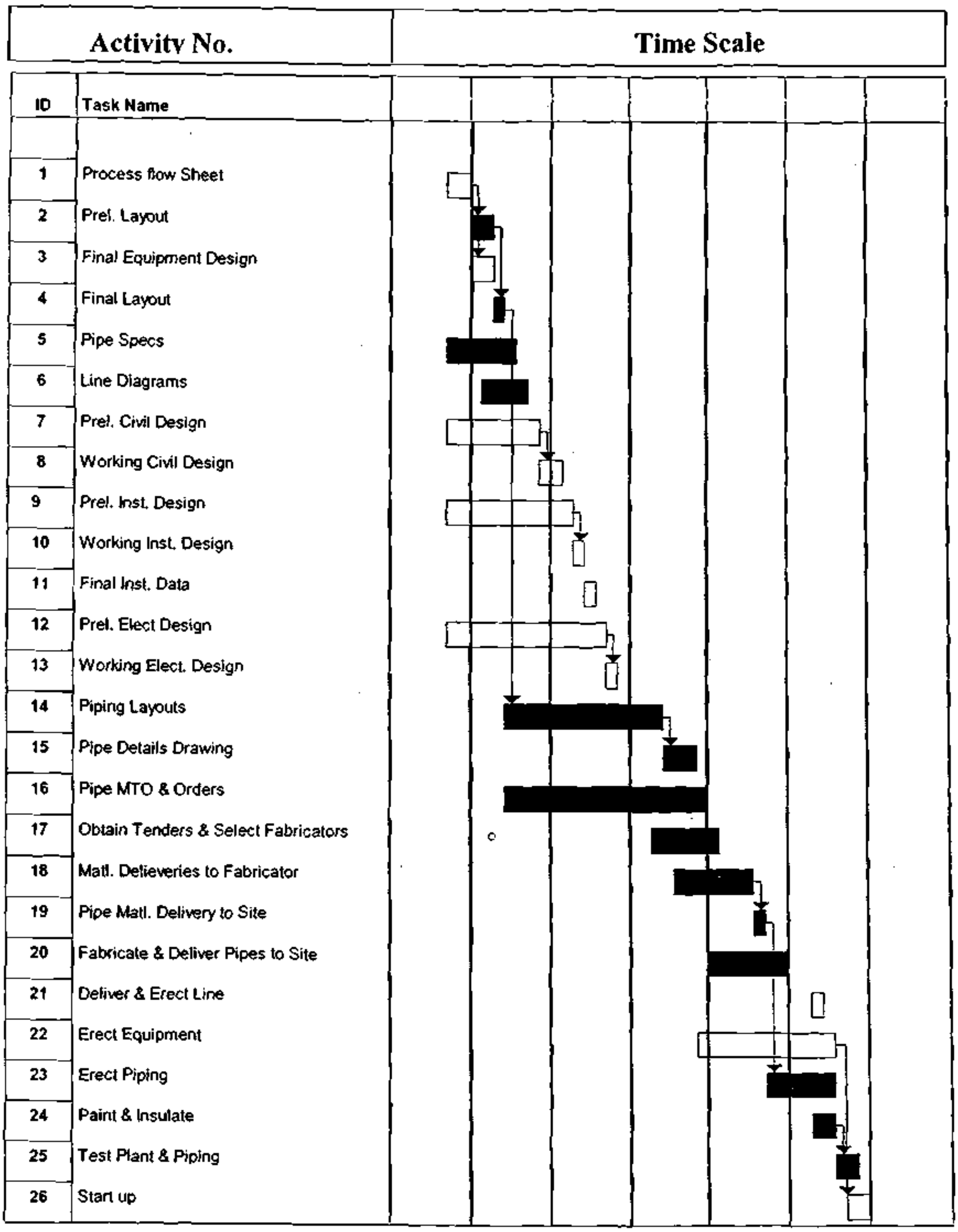


FIG. 6 BAR CHART FOR TYPICAL PROJECT ACTIVITY

(Piping Activity Shaded)

PIPING SECTION

	INPUTS TO PIPING			OUTPUTS FROM PIPING
1	Data as per project design basis such as indicative Plot Plan, PFDs, P&IDs, PDS, Process description, Equipment list, Line list, Site data, etc.	-- Process licensor	1.	Overall plot plan showing location of various units, tankfarms, offsite, package units, non-plant buildings, roads, culverts, piperacks, sleepers, etc.
2	Electrical tray width requirements on piperack/pipe sleepers and cable trench width in units/offsites	-- Elect.	2.	Piping material specifications.
3	Instr. cable tray size requirement on piperack/sleepers	-- Instr.	3.	Equipment layouts (1:200), (1:100), & (1:50).
4	Engineering data sheets of equipments like columns, vessels, tanks, etc.	-- Mech.	4.	TOG Elevations and loads with anchor bolt size, no., and bolt locations for all equipment.
5	Engg. data sheets giving overall dimensions, supporting arrangement with no. of anchor bolts, size, location and bolt circle dia.. All nozzles location, size, rating, etc. for heat exchangers.	-- Process	5.	General arrangement of piperack and equipment supports structures including loads.
6	HVAC-ducting layouts preliminary and final.	-- Utility	6.	Piping general arrangement drgs. including platforms, ladders, overhead cranes elevation, location, loads and layout of monorail, cutouts, inserts and sleeves, required for piping, etc.

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7	Layouts of various effluent & drain sewers and location of manholes.	-- Civil	7.	BOM & technical evaluation and technical bidding analysis.
8	System preliminary and final layout drawings of the packaged units including auxiliary equipments.	-- Utility	8.	Stress analysis
9	Architectural drgs. of instr. control room, electrical sub-station, laboratory and other offices inside the unit-plans and elevations.	-- Civil	9.	Nozzle orientation drgs. including location of davits.
10	GA drgs., overall dimensions, base plate and anchor bolt locations, nozzle locations, size, type, rating, type of drive, permissible nozzle loads, etc. for pumps.	-- Process	10.	Vessel cleats location drgs. for pipe supports, platforms and ladders.
11	Foundation and super structure drgs. of piperacks and process/technological structures.	-- Civil	11.	Underground piping layout.
12	Overall foundation layout.	-- Civil	12.	Isometrics of piping including system ISOs
13	Platform drgs. and special pipe structure support structure drgs.	-- Civil	13.	Support drgs. of piping and special support details.
14	General arrangements drgs. of electrical cable tray/trench layouts.	-- Elect.	14.	Tender for fabrication S.O.Q.

15	Tank settlement data.	-- Civil	15	Tender for insulation S.O.Q
16	Size and rating of control valves/safety valves connecting flanges.	-- Instr.	16	Tender for painting S.O.Q
17	Sketches for level instruments (nozzle elevation).	-- Instr.		
18	Dimensional drawings of all inline instruments.	-- Instr.		

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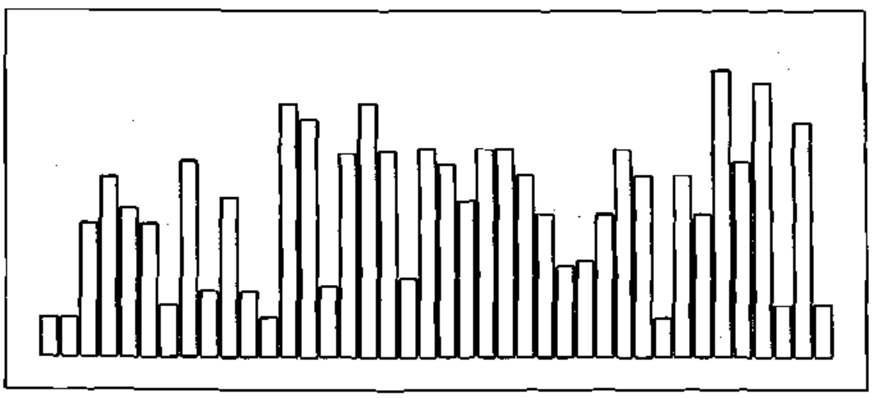
PIPING ENGINEERING
A Major Phase in the Life Cycle of Process Plants

Prof. A. S. Moharir
IIT Bombay



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PIPING ENGINEERING: A MAJOR PHASE IN THE LIFE CYCLE OF PROCESS PLANTS

PROF. A. S. MOHARIR
Indian Institute of Technology

INTRODUCTION

The life of a chemical process, from concept to commissioning and beyond, involves almost all disciplines of engineering. So wide is the knowledge base requirement and so intricately integrated these inputs from various specializations are that they make the conventional engineering disciplines such as chemical engineering, mechanical engineering, metallurgical engineering, civil engineering, etc. look artificial. A good chemical process engineer needs to have a very broad knowledge derived from these disciplines.

The idea of this paper is to take a bird's eye view of the activities during the life cycle of a process, especially those that concern a piping engineer.

MAJOR PHASES

The major phases in the life cycle of a chemical process can be identified as:

1. Determination of Techno-economic feasibility
2. Design Phase
3. Construction Phase
4. Commissioning Phase
5. Operation/Production Phase

A piping engineer has an important role to play during phases 2-5.

1. TECHNO-ECONOMIC FEASIBILITY

Except perhaps in the early days of metallurgical industry, economic gain has been a major consideration in the choice and scale of a production activity. Risk factor and pollution considerations seem to be gaining importance, but these are also due to economic considerations to a large extent, because ignoring these may lead to penalties in future as harsh as having to top the activity. So, economics remains the guiding principle. There is nothing to wrong with it because such a profit motive has led to great technological innovations.

For new products, technical feasibility is an important first aspect to study. This can comprise of

– Chemical Path Feasibility
– Engineering/Technological Feasibility

For the first part, especially in the area of organic synthesis, reaction path synthesis algorithms are available. These would short-list a few routes for synthesis which are thermodynamically feasible. Every thermodynamically feasible reaction need not necessarily be practical because for practicality, it is essential that the reaction takes place at a suitable rate. Too fast reactions may lead to operation/control problems while too slow reactions may require huge equipment to be able to process commercial quantities. Means of promoting/inhibiting reaction rates(catalysts) are often required. A major portion of reactions in chemical industry are catalytic in nature. A general view of any chemical process plant can be a reactor at the heart and other units necessary to prepare feed for the reactor or process output from the reactor.

Once a chemical route which is thermodynamically and kinetically feasible is chosen, a block flow diagram(BFD), which can be said to be the first engineering diagram in the life cycle of a process, can be prepared. It simply shows the operations that are involved in the feed preparation section

process plants are generally designed for 20 yrs

and product treatment section of a process and the general sequence of events. The operations could be heat removal or addition, mixing or separation, pumping/compressing, etc. For a given scale of operation and estimated or expected performance levels of the equipment to be later selected for these operations, one can put first estimates of stream characteristics such as flow rates, composition, temperatures, pressure etc. on the BFD. With this, the process can be said to be born in two dimensions.

The chemically feasible route is yet to be tested for its technical and economic viability.

At a broad level, a decision as to the desirability of batch operation, continuous operation or a combination has to be taken. Scale is an important issue here, but not the only one. Some operations are inherently batch/semibatch in nature, e.g. adsorption. The portion of the flowsheet involving batch operation must operate in that mode. It can suitably coexist with upstream/downstream continuous operation through provision of suitably sized storage tanks which do the job of isolating the batch operation section from the continuous one.

Some operations are feasible in batch as well as continuous mode but strict quality control (absence of byproducts due to side reactions etc.) or variations in feed/products specifications etc. may tilt the scale in favour of batch operation. For the same scale of operation, batch process equipment result in much larger process fluid inventory in the plant at any time. If the fluids being handled are hazardous, it would mean that potential hazards are higher in batch operation vis-à-vis continuous operation. This aspect is assuming more and more importance in the emerging zero-risk scenario.

Once the operations and the mode in which these are to be carried out are determined, the actual methods of achieving results of an operation are to be decided. The total mass is obviously conserved in any process including the chemical processes. All operations in chemical processes wherein the masses of individual components are also conserved are termed as unit operations. This would necessitate that no chemical transformations take place during these operations. Phase change, is however, not excluded. Pipe flow, pumping, compression, mixing, evaporation, distillation, extraction, etc. are unit operations in this sense.

The heart of any chemical process, a reactor, is however, an operation where the total mass is conserved but the species mass is not. Some or all the species undergo chemical transformations. These are called unit processes. For example, nitration, oxidation, hydrogenation, chlorination, esterification are called unit processes.

The operations in a process as identified in the BFD may be carried out by one or more candidate unit operations. For example, a component from a mixture may be separated by distillation or crystallization or adsorption. Which choice is the best (from economic point of view)? Is a parallel or series combination of alternatives a better solution?

Even after selecting the appropriate unit operation, one may have to converge on the appropriate implementation strategies. For example, a four component mixture may be separated by multiple distillation columns in several possible ways. Which one of these options is the best for a given situation?

In case of reactors, choice of equipment is equally important. A fluid phase reaction, for example may be carried out in a tubular reactor with or without recycle, a stirred pot or a combination. A reaction involving solid catalyst may be carried out in a packet bed, moving bed, basket type reactor, a fluidized bed reactor, a riser reactor, etc. Which one of these options is the best for a given situation?

1st thing to separate something is distillation.

batch process involves a lot of inventory while continuous process doesn't involve inventory much

Plant cost and operating cost are two components to be considered in the choice of equipment. It is possible to pose the problem as an optimization problem (cost minimization), the solution of which gives cost optimal flowsheet configuration. This phase of flowsheet development is called "Process Synthesis" and widely accepted CAD tool in chemical engineering. Mathematically, these are MILP (Mixed Integer Linear Programming) or MINLP (Mixed Integer Non-Linear Programming) problems.

Selection of other pieces of equipment such as pumps, reboilers, heat exchangers can be taken up at and considered as a part of process synthesis stage or postponed to a later stage.

Chemical processes are generally energy intensive. With the rising energy cost, attention has recently turned to minimization of external energy requirement. These are cooling water, steam, heating oil etc., the so called utility steams. It is essential to minimize utility requirement by encouraging as much process stream to process stream heat transfer as possible within operational and layout constraints. These problems are also mathematically posed as MILP and MINLP and are called as HEN (Heat Exchanger Network) Synthesis problem.

With these process synthesis tasks accomplished, conceptual design stage of the process is over. The equipment have been selected and roughly their capacities are known. The capital cost and the operating costs can be estimated. This information coupled with the raw material availability and cost and the demand and market prices of products and byproducts would help in establishing the economic feasibility of the process. A techno-economically feasible process is now ready to enter the design phase.

2. DESIGN PHASE

The objective of the design phase is to carry out rigorous engineering calculations for the chemical engineering aspects as well as mechanical engineering aspects and come out with rigorous documents (text, drawings) so that the implementation details can be passed on to the subsequent phases in the life cycle.

It must be remembered that just as it is important to know what is happening inside a particular piece of equipment(chemical engineering), it is equally important to know whether the mechanical design and metallurgy of the confining vessel are adequate to allow this to happen without risks to the other flowsheet components and to the external world. The issues are inseparably involved and chemical and mechanical engineering have to go hand-in-hand. Unfortunately, that is normally not the case.

The design phase has essentially two components; the process design and the mechanical design.

Process is the detailed material and energy balance calculation across the process flowsheet. It would also establish the operating conditions, equipments size(not necessarily shape), utility requirements, etc. With this information appended to the BFD, one gets a Process Flow Diagrams(PFD).

For example, process design of a distillation column would mean calculating the number of trays, feed tray locations, draw locations, draw locations, condenser and reboiler duties, reflux ratio, etc. The heat exchanger process design would mean calculating the operating temperature, pressure, reactor volume, reactor heat removal/addition requirements, etc. These operating/design conditions can be calculated to satisfy some criterion such as minimum operating cost or minimum impurities or minimum pollution or maximum conversion to desired cproduct or a combination of these.

In PFD, symbols shd be shown, except the packaged units for which blocks are used.

These decisions which help to zarrive at optimal design and not only workable designs are possible only by creating and answering what-if situations around each equipment or part or whole of the flowsheet. For example, we need to try different feed locations for a distillation column, different reflux ratios etc. and see how the steady state performance(top or bottom composition) changes. A feed location and/or reflux ratio which gives the desired performance is one of the candidate designs. There may be several such designs. The one which is optimum in some sense is then chosen. Prediction of performance for inputed design and operating conditions by solving phenomenological equations of any operation on computer is called simulation. Programs which can do it for the whole flowsheet simultaneously are flowsheeting programmes. They are also termed as steady state simulators because they simulate only the steady state performance of a process flow sheet.

Simulation based process design completes one very important phase in the life cycle of a chemical process. The equipment types and sizes, all stream specifications (Flow rate, composition, temperature, pressure), operating conditions are known at this stage. This, when incorporated in Bfd, converts it to a PFD, an important engineering diagram. It is still 2-D, but since it has significantly more information content, let us call it a 21/4-D drawing. This is often considered as an end-product of a conventional chemical engineer.

Although widely used in process design, steady state stimulators do not help in decision making during several important stages of a chemical process. The important ones start up and shut down, transition phase during feedstock and product changeover, relief and blow down, control systems synthesis and design, Hazard and Operability (HAZOP) studies, etc. These situations require the knowledge of dynamics of a process. Dynamic process simulators are being developed and slowly getting acceptance as decision making tools in these areas.

Although designed for steady state operations, truly speaking, no process operates at a steady state. This is because there could be disturbances (may be with zero mean) beyond the control of operators. Even if we have a tight hold on everything, the ambient conditions (temperature, wind velocity) change from time to time. This changes the amount of heat ingress or egress from the equipment and pipelines which affect the energy balance of the whole system and system performance would vary if no counter measures are taken. This is the job of suitably chosen controllers.

Control system design is an important area. System identification, modeling, manipulated-controlled variable pair selection, controller selection, controllability evaluation are important areas. Dynamic simulation packages need to be used for this purpose so that CSSD can be done at process design stage.

HAZOP is another important and now mandatory activity. It is a qualitative, experience intensive exercise as of now. It is in the form of deviation analysis. After the process design, the steady state specifications of each stream in the flowsheet are known. The HAZOP team exhaustively asks itself questions as to what will happen if these specification deviate from the expected steady state value. It debates the possible causes and consequences of each such eventuality. Anything which appears to them as likely to lead to hazardous situation is debated further and possible means of avoiding the same or raising alarm if it happens so that remedial action can be taken etc. are recommended. This may lead to recommendation of additional instrumentation on lines and

equipment, high-low alarms and trips etc. which may be required to be provided.

The idea of HAZOP is to foresee hazardous situation and take measures and abundant precaution to avoid them and increase process safety.

The requirement of monitoring instruments for providing signals to controllers or also to continuously monitor process performance is identified.

The PFD shows the flow rates, composition, temperature and pressure of all feed, product and intermediate streams. The properties of these streams such as density and velocity can therefore be calculated. Hydraulic calculations to decide the pressure drop due to flow from one unit to another can be done at this stage. Pipe sizing which is a balance between operating cost (energy lost due to flow in pipes) and capital cost (function of pipe diameter, thickness and pipe run) can be carried out.

The pressure drop correlation to be used depends on the nature of flowing medium: incompressible, compressible, slurry, two-phase, three-phase, etc.

Reasonably good correlation are available for calculating pressure drop for single phase compressible or incompressible fluid flow. For two-phase flow, correlations are available but their predictive power is doubtful. Possibility of various flow regimes, uncertainties regarding regime transition boundaries and lack of data make prediction flow hydraulics very difficult for two and multiphase flow. All one can do is to use the best available correlation. Unfortunately, a large percentage of flow situations in industry are atleast two-phase flows.

The HAZOP findings, process requirement or the available reliability data may require a standby unit provision in the flowsheet. This is quite often the case with pumps. These call for appropriate piping also.

To make sure that a batch or semi-batch operation can coexist with upstream and/or downstream continuous units, storage tanks need to be provided as was mentioned earlier.

Sometime, a particular requirement of an equipment may call for such a provision, for example, to ensure that the pump suction does not run dry in the event of upstream process upsets. Start-up or shut-down conditions may also call for intermediate storage tanks.

A PFD modified further to indicate stand-by equipment, storage tanks, instrumentation and control, pipe sizes, valves, etc. becomes a P&ID. It may also show relative elevation of various equipment, number of trays and feed tray location in a distillation column, etc.

P&ID is a very important schematic during the design stage process. In fact this is what is stored and updated throughout the life of a process. It is considered as a cardinal drawing for various sections in a design organization. It also is the basic drawing for subsequent equipment design, plant layout, piping layout, bill of material(BOM), insulation calculations, etc. Project engineering literally begins with this mother drawing in hand.

Piping Engineer must be thoroughly conversant with P&ID. Each organization may have its own nomenclature and practices for making P&IDs. But the differences are mostly in representation and not in the information content.

P&ID also shows the other details of the pipe lines such as material of construction, service, etc. The material of construction can be decided based on the fluid that a pipeline is supposed to carry and the temperature-pressure conditions. The materials of construction for equipment similarly are service dependent.

Although a piping engineer begins with P&ID, it would be desirable if he has

some knowledge of the process background which led to P&ID.

At this stage, the capacity of each equipment, the temperature- pressure that it needs to withstand, the material of construction, the inlet-outlet ports and their sizes that need to be provided, the necessary details of other internals/externals(such as trays in a distillation column, packing in an absorber, stirrer in a reactor, jacket around kettle etc.) are available. The detailed mechanical design of each equipment leading of fabrication drawing can be taken up.

The equipment design falls in the area of pressure vessel design. The pressure vessels are classified as fired and unfired depending on whether they come in contact with naked flame or not. Vessels subjected to inside pressure higher than the ambient(eg. vacuum service) are to be designed separately. There are separate design formulae for vessels subjected to internal and external pressure.

Typical vessel shapes used in process industry are cylindrical and spherical. Spherical shells are self-closing while two ends of a cylindrical shell need to be closed using closures of appropriate shapes. Depending on shell dimensions and service conditions, the closures may be hemispherical, ellipsoidal, torrisperical, conical or flat.

The closures need to be tightly fixed to the shell. Flanges are provided on the closure and the shell for this purpose. Various types of flanges are possible and appropriate choice is important. Ingress of ambient air in the vessel or egress of inside fluid to the atmosphere must be avoided. Apart from loss of material and/or off-spec product, such a leakage could be hazardous. If flange surfaces were pressed against each other, no matter how well they are machined, the flanges would leak. This is so because of the microscopic irregularities on the flange surface, which act as channels for leakage. These channels need to be blocked by providing a softer material which is squeezed between the two flanges so as to flow and seal the irregularities on the flange surface. The gasket must flow but not be squeezed out of flanges during bolting up conditions. Also during operation, some of the bolt tension and pressure on gasket is reduced. The gasket should still not leak.

The pressure vessel design would involve calculating the shell wall thickness, closure type selection and thickness calculations, selection of suitable gasket material with adequate yield stress and gasket factor, choice of gasket location(mean gasket diameter), gasket thickness, gasket width, placement of bolts (bolt circle diameter), bolt material selection, number of bolts, diameter of bolts etc. But this is not all.

A vessel needs to have openings to serve as inlet, outlet ports as well as for drainage, hand holes, man holes, etc. these could be on the shell or closures. The shell and closure wall thickness have been designed to ensure that the stresses in the walls even at the weakest ports (along welding, along longitudinal seam or girth seam) do not cross the allowable stress value even after corrosion or inspite of nonuniform plate thickness (mill tolerance), etc. When openings are cut, stresses concentrate along the edges of the opening and may exceed allowable stress value. Provision of extra thickness to counter this may be expensive. The theoretical finding that the stress concentration is confined to a circle double the opening diameter is used to strengthen the shell wall only in that region by welding a pad around the opening. The thickness calculation of such a reinforcing pad is a part of pressure vessel calculation.

Not every opening need to be compensated. In the pressure vessels subjected to internal pressure, tensile stresses

are developed in the wall. In case of vessels subjected to external pressure, compressive stresses are developed. The vessel wall would have a tendency to buckle. To avoid this, stiffening rings may be provided. Spacing between stiffening rings and cross-section geometry of the stiffening ring are to be designed using appropriate design procedures.

Stiffening rings may be provided externally or internally. Internal stiffening rings may also be used as tray supports in distillation columns, etc.

Above design procedures may normally be adequate for not so tall vessels. For tall vessels which may most adequately be called towers, several other considerations come into picture.

Tougher distillations require lot of trays in the column requiring to install very tall columns. Tall column design is thus important.

The wind velocities increase as one goes away from ground level. A tall column with its insulation, platforms and ladders provide obstruction to wind which in turn exerts force on the column. The column firmly supported at the skirt top bends as a result. This induces tensile stresses along the longitudinal seam on the leeward side. These additional stresses along with the stresses due to internal pressure should not cross the allowable value. This may have to be ensured by provision of additional shell wall thickness. The thickness requirement is lowest at the top. To minimize metal requirement, the tower may be divided into sections with the bottommost section having highest thickness and the top just enough to withstand internal pressure.

Tall tower design also needs to be checked for seismic effects which induce additional stresses along longitudinal seam. The seismic zone and the period of vibration decide the seismic coefficient. Wall thickness may have to amended to take care of seismisity.

It is assumed that highest wind load and worst seismic effects do not occur simultaneously.

The vibration period and deflection of tall columns also need to be kept within tolerable limits. This can be done by providing a suitably thick skirt.

Eccentric loads on the column due to side connection also cause bending moment at the skirt column connection and need to be considered.

After attention is paid to each and every aspect, a pressure vessel fabrication drawing is issued for fabrication to begin at an early stage. Equipment fabrication is time consuming. Also, in the field work, equipment need to be in place quite early because only then the pipe routing job begins.

Pipe wall thickness design is similarly carried out by treating pipes as cylindrical vessels. Flange calculations need not be done in the case of pipes as these are provided by the codes in most cases. Once the nominal diameter and schedule of all pipes in a plant are known, a first bill of material for pipe length requirement can be prepared. This is even before the actual routing and isometrics are frozen. Additional quantities can be procured once the 3-D layout is finalized.

In all the above calculations, design pressure and design temperature have to be suitably chosen. For the equipment, hydrostatic test pressures are also to be recommended.

With the completion of these calculations and the design documents, the design phase can be said to be over. The pipeline routing is however not yet decided.

3. CONSTRUCTION PHASE

Further analytical work needs to be

Site selection
plot plan / layout
Unit/equipment layout

done before the final blue print of 3-D plant layout is finalized and construction begins. Some of the activities given here may well be considered as belonging to the design phase itself. They are given here mainly because plant site details are a part of inputs to the decision making.

The choice of plant location, if such a choice exists, is governed by politico-socio-economic considerations. The basic approach is to assign weight factors to various considerations and to select a site which scores maximum points.

Knowing the site and its neighborhood, a plant layout can be worked out. Sites of the major equipment have to be decided on the site map. Apart from the equipment and offsites, other requirements such as control room, fire station, hospital etc. are decided at this stage. The road map of the site also emerges. Certain rules for inter-unit distance which emerged from past experiences, certain guidelines for dusty, fire-prone, noisy, hazardous equipment location are adhered to.

A piping engineer is deeply involved in plant layout as it is one of the most important factor which governs the piping layout and piping costs.

It is important to orient the unit properly at it's assigned site. This activity is called unit layout or equipment layout. Accessibility, ease of maintenance, implications on piping layout, etc. are the considerations here. Each equipment has to be given individual attention by always keeping in mind that it is a member of the whole. Guidelines have emerged based on past practices and experiences.

A piping engineer is again deeply involved in unit layout as it has more direct influence on the piping layout, which is the next activity.

After the units have been located and appropriately oriented on paper, the layout of the veins and arteries of the plant, the pipes, have to be laid out. It is not as simple as connecting the outlet of one equipment to the inlet of the next in operational sequence by the shortest possible route. In fact, such direct connections are exceptional. And with reason!

Most industrial hazards originate with failure of the piping system. The equipment are fairly rigid and have strong foundation. During the cold assembly, all pieces are in place. When the operation begins, the high pressure temperature conditions inside the equipment induce stresses and things literally move. Pipelines being the most delicate elements in the plant, bear the brunt of these operational loads. It is therefore essential that each pipeline routing, especially the critical long ones, is properly designed so that the pipe can sustain the operational load. The load due to the weight of the fluid carrying pipe, vibrations in the equipment to which it is connected, thermal expansion etc. collectively and should not lead to stresses in the pipe exceeding the allowable limits during operation.

Weight analysis and stress analysis need to be carried out on pipeline. It may lead to the requirement of rerouting the pipeline, or provision of supports, hangars, expansion bellows, etc. stress analysis is now facilitated by computer packages. However, analysis of the stress distribution churned out by these packages for a complex pipe routing is the job of a piping engineer. Piping layout is an exclusive domain of a piping engineer. Not many get exposure to it during their career as piping engineer.

A software model of 3-D layout of a plant is gaining imporatnce. Unlike P&ID which is schematic, a 3-D model is a dimensional graphics and can be made in all details of the envisaged plant. It offers easy visualization of the plant structure. It allows checks on ergonomics. In conjunction with stress analysis software, each pipeline can be checked for adequate flexibility and its route

if called for. Details of civil structure can be checked and corrected. Isometric drawings of each pipeline can be derived from 3-D drawings with ease. Orthographic drawings in different views can be created.

The 3-D software model has all the details of plant including actual pipe routing. Bill of material for pipes and piping elements (pipe run, piping elements such as elbows, tees, specialties, flanges, valves etc.) procured equipments (pumps etc.) can be easily extracted.

The progress of project implementation can be monitored using 3-D drawings and field information.

Specification sheet for piping elements can be prepared using a 3-D model.

3-D model of the plant is complete database and visual of the plant that would be. Its use would increase in coming years.

The construction phase involves the actual placement of equipment and routing pipelines. Welding and fabrication, painting for corrosion prevention, thermal insulation to prevent heat ingress or egress are the field activities that a piping engineer need to be familiar with.

4. COMMISSIONING

If the entire design has been done scientifically, if design intentions are reflected in various design documents correctly, if fabrication, erection and assembly have been done as per design intentions, then commissioning which involves taking the cold-assembled plant to go on-stream and produce design capacity should be smooth affair. This is normally not the case because lots of adhoc decisions need to be taken on field during erection to take care of fabrication errors, late or non-delivery of items or design errors which are made at early stages of project engineering or even late second thoughts. The project is normally on the critical path during field work and not all these decisions and their implications are thoroughly probed.

Another reason why commissioning is tough is that the start-up conditions are significantly different than steady state conditions for which the plant has been designed.

Dynamic simulation is a good tool to evolve a good start-up policy. It is, however, not used much of today. Start-up procedures for common unit operations such as distillation are fairly well tested, though not necessarily optimal.

A piping engineer may be involved in star-up to take care of mechanical design problems that may crop up. Some process knowledge would be desirable.

5. NORMAL OPERATION PHASE

The problems during the production phase of plant are mostly operational if it has been designed well. The need to debottleneck and optimize on throughput, however, calls for minor/major changes involving installation of additional equipment or bypassing an existing equipment and related changes in pipe routing. These changes may be trivial from process point of view but not necessarily from the mechanical design implications point of view. A trivially simple change may lead to stresses crossing failure limits and causing disaster. A healthy operating practice would require a piping engineer to be associated with any hardware change or operating point shift during productive part of the life cycle of a plant.

A piping engineer is also a part of HAZOP team. He is also involved in accident review.

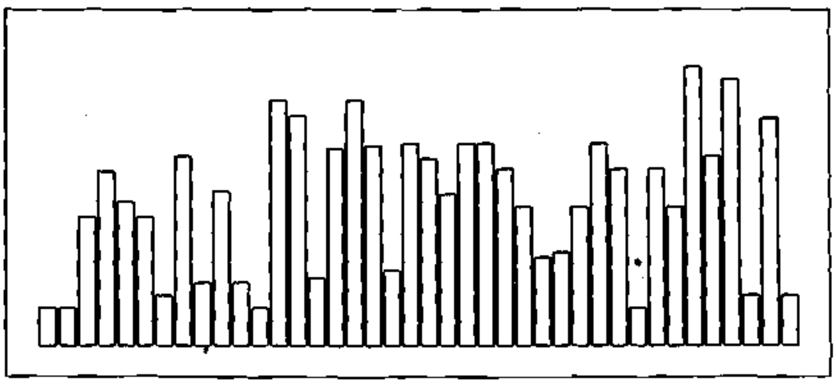
IN CONCLUSION, A PIPING ENGINEER IS INVOLVED I A MAJOR PORTION OF THE LIFE CYCLE OF A PROCESS. HIS RESPOSIBILITIES AND SCOPE OF ACTIVITIES MAKE THE NOMENCLATURE "PIPING ENGINEER" A MISNOMER.

Certificate Course
on
PIPING ENGINEERING

June 12 - 25, 2006

PIPE HYDRAULICS AND SIZING

Prof. A. S. Moharir
IIT Bombay



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PIPE HYDRAULICS AND SIZING

DR.A.S.MOHARIR
DR.P.PARANJAPE

WHY PIPE SIZING IS IMPORTANT

According to a 1979 American survey, as much as 30% of the total cost of a typical chemical process plant goes for piping, piping elements and valves. A significant amount of operating cost (energy) is also used up in forcing flow through piping its components. A significant amount of the maintenance cost is also for the piping and associated things.

Proper sizing, optimal in some sense, is therefore very necessary.

WHY IS IT DIFFICULT AND AT TIMES MEANINGLESS

Piping must be sized before the plant is laid out. Layout must be complete (i.e. equipment must be located, pipe racks established, layout of individual pipe runs decided, etc.) for calculating realistic pressure drop and doing pipe sizing for each pipe segment. This 'chicken and egg' scenario means that decisions regarding pipe sizing and plant layout must be iterative in most cases. That is normally not the practice except in few very large engineering organizations which can afford it. Having to carry out pipe sizing at a premature stage invariably means that the recommended pipe size may not meet process requirement or may not be the most economic, etc.

Normally a layout is assumed drawing on past practices and experience and pipes are sized. No second iteration is carried out. Actual layout which emerges later may be significantly different than what was assumed during sizing. The sizes thus may turn out to be non-optimal. Also, what is optimal today may not be optimum over a long period (due to fouling, change in relative cost, change in operating schedule which affects the utilization time of the pipeline,etc.)

Pipe sizing is thus a lot of experience, engineering foresight and judgment than just theory. This paper attempts to review the pipe sizing procedures, the pressure drop calculation procedures which are integral to pipe sizing procedure, the pitfalls in these calculations, the confidence limits in calculated values and the factors of safety which must be incorporated in view of known limitations of correlations. Different concepts are then cemented through representative examples during the lecture in the Certificate Course on Piping Engineering conducted by Piping Cell at Indian Institute of Technology, Mumbai.

PIPE SIZING PROCEDURES

Pipe sizing is generally done using one of the following criteria:

1) Velocity considerations
2) Available pressure drop considerations
3) Economic considerations

The degree of difficulty increases as one goes from (1) to (3). While pressure drop calculation in an integral part of (2) and (3), it would need to be calculated in case (1) also to quantify energy requirement, sizing pressure providing equipment such as pumps/ compressors, etc. To be conversant with pressure drop calculation procedures for variety of flow types that are encountered is thus very important.

This paper assumes that the readers are conversant with pressure drop calculation procedures and concepts underlining them, at least for the single phase flow. The paper attempts to build on this background.

The paper reviews the following:

TYPES OF FLOW

- Single phase, Two phase, Multi-phase
- Horizontal, Inclined
- Through straight run-pipes, through complex routings
- Isothermal, non-isothermal
- Incompressible, compressible
- Laminar, Turbulent

BERNOULLI'S EQUATION

SINGLE PHASE PRESSURE DROP CALCULATIONS

- Horizontal, straight, constant cross-section segment
- Inclined, straight, constant cross-section segment
- Fittings and valves
- Equivalent length in actual terms
- Equivalent length in diameter terms

TWO PHASE PRESSURE DROP CALCULATIONS

- Flow regimes and their identifications (Baker Parameters)
- Pressure drop calculations (Lockhart Martinelli, Baker)
- Confidence levels in calculated pressure drops
- Effect of inclination
- Scientific approach

MULTI CALCULATIONS -PHASE FLOW PRESSURE DROP

- A possible approach

PIPE SIZING

- Velocity considerations
- Pressure drop considerations
- Economic considerations

TYPE OF FLOW

Although the flow can be categorized on several basis the classification based on number of phases involved is the most commonly used. When the flowing medium has uniform physical properties across the flow cross-section, the flow is a single-phase flow. Flow of pure single liquids, solutions of solids in liquids, mixtures of completely miscible liquids, mixtures of gases and/or vapors come in this category.

All other flows are multiphase flows. The two phase flow would involve two distinct phases such as liquid with its vapor, a liquid with an incondensible gas, etc. A liquid or gas/vapor stream with suspended solid particles is also a two phase flow. However, a two phase flow would normally refer to two fluid phases. When two immiscible liquids are involved with their vapor and/or another inert gas, it is a three phase flow and so on.

Energy required to sustain such flows in pipes/tubes is a very important information which has to be generated through calculations of pressure drop that the flow would cause in a conduit of given cross-section, and extent. This information is then used in locating equipments, sizing

Single phase → at the bottom & top, same fluid is flowing.
(any of vapour, gas, liquid)
→ properties of fluid is same all over the

pipes, deciding their routes, rating pressure generating equipments, etc.

Temperature of the flowing medium affects physical properties such as density and viscosity which in turn have a bearing on the pressure drop. When the temperature is constant over the pipe segment under consideration, or the temperature change along the flow path is not significant enough so as to cause appreciable change in the physical properties, it is treated as an isothermal flow. When the temperature change is significant, it is non-isothermal flow. When the density of the flowing medium is not strongly correlated with the pressure, the medium is termed as incompressible and the flow as incompressible flow. Liquid flow (single, two or multiphase) would come in this category naturally. However when gases/vapors which are compressible (that is their density is a strong function of pressure) are involved, but the pressure drop along the flow path is not significant enough to affect the medium density, their flow may also be treated as incompressible flow. Otherwise, the flow of gases/vapors is a compressible flow.

In some flow situations, especially two and multiphase flows, the inclination of the flow conduit from horizontal is of great significance. Also whether the flow in the inclined conduit is upward or downward is also an important consideration. In the case of single phase flow, the inclination is important in the sense that it affects the overall energy balance given for the flow situation by the famous Bernoulli's equation. But the flow type and hydraulic pressure drop are not affected by the pipe inclination.

BERNOULLI'S EQUATION

In its original form, Bernoulli's equation is merely statement of conversation of energy for flowing medium. Consider a segment of an inclined conduit of variable cross-section as shown in Fig.1 and fluid flowing through it. The energy of the fluid at any location may be expressed in terms of a vertical column of the flowing fluid itself. The height at any point along the conduit is then seen as comprising of three components, the pressure (P/ρ), velocity head ($v^2/2g$) and elevation head (Z). Bernoulli's theorem states that the sum of these three components is constant everywhere along the flow path. This is true if there are no external inputs or withdrawals from the conduit Applied at the two points 1 and 2 of the inclined pipe shown (Fig.1), the Bernoulli's equation can be written as follows: (energy of the fluid during the flow is same)

$$P_1/\rho+v_1{}^2/2g+Z_1=P_2/\rho+v_2{}^2/2g+Z_2$$

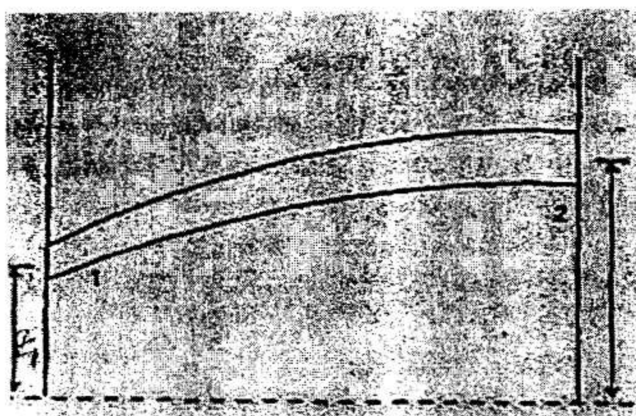


Fig.1

When the pipe is horizontal ($Z_1= Z_2$) and the conduit cross-section is uniform ($v_1=v_2$), the pressures at the two points, 1 and 2, should be equal, This is not the case because the flow is confined by the pipe and there is a resistance to flow caused by friction between the fluid and the wall, friction between different layers of fluid flowing at

- 2 phase pressure drop is very high

different velocities and the small or big swirls created in the liquid due to flow turbulence. Flow against these resistances causes generation of heat raising the temperature of the fluid as it flows. This temperature rise is not enough to do any work an this energy transformed into thermal energy is good as lost energy. This expressed in pressure units or expressed in terms of an equivalent column of the flowing fluid is called frictional pressure drop or head loss.

Incorporating this fact into the Bernoulli's equation yields the following form which is generally used in calculating frictional pressure drop in flow:

$$P_1/\rho+v_1^2/2g+Z_1=P_2/\rho+v_2^2/2g+Z_2+\Delta P/\rho$$

SINGLE PHASE PRESSURE DROP CALCULATIONS

Single phase flow is classified as LAMINAR, TRANSIENT OR TURBULENT. The deciding factor is the REYNOLD'S NUMBER defined as follows:

$$R_e=\frac{Dv\rho}{\mu}$$

It is a Dimensionless number if the quantities are in consistent units. For Reynold's number values up to 2000, the flow is termed laminar and for values above 4000, it is a turbulent flow. The range 2000-4000 is termed as the transition region. D in the definition of the Reynold's number is the actual diameter if the flow cross-section is circular such as in commonly used pipes.

v = vol. flow rate / area

However, for other cross-section (rectangular, square, annular, etc.), D is defined in terms of the Hydraulic radius(R_H) as follows:

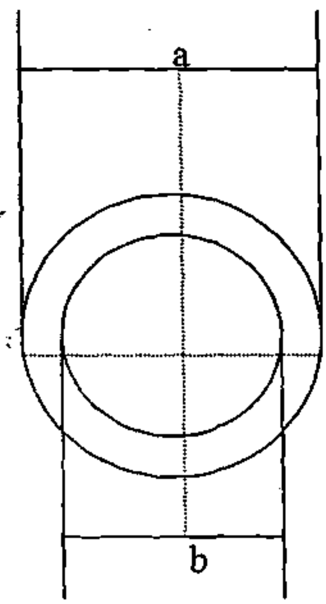
D = 4 x Hydraulic radius

The HYDRAULIC RADIUS is defined as ratio of flow cross-sectional area to the wetted perimeter. For example, in the case of a rectangular cross-section with sides a and b, the flow cross-section is ab while the wetted perimeter is 2a+2b. Similarly, for an annular region as shown (Fig.2), the hydraulic radius is as shown:

$$R_H=\frac{[\pi(D_2^2-D_1^2)]}{[4\pi(D_2+D_1)]}=\frac{(D_2-D_1)}{4}$$

D_2 = inner dia of outer pipe

D_1 = outer dia of inner pipe



With D defined in this general sense in the definition of Reynold's number, the limiting values of the number for laminar, transient and turbulent flows remain the same as given earlier. The linear velocity used in the definition of Reynold's number is obtained by dividing the volumetric flow rate by cross-sectional area for flow.

density calculation
sweet water

Viscosity is not so imp in turbulent flow.
101.325 Kpa = 1.033 Kg/cm^2 ≈ 1 atm

Alternative but equivalent forms of definition of Reynold's number which are commonly used are as follows:

$$R_e = \frac{DG}{\mu}$$

Where G is the linear mass velocity of fluid

$$R_e = 6.31\frac{W}{(D\mu)}$$

Where W is the mass flow rate lb/hr, D is pipe ID in inches and ρ is density in lb/ft^3

The frictional pressure drop is calculated using Darcy's equation as follows:

mass flow rate

$$0.0000226\frac{f_D q^2}{D^5 \rho} = \boxed{\Delta P/\rho = \frac{(f_D v^2)}{(2gD)}} \quad 0.810\frac{f_D q^2}{\rho g D^5}$$

f_D is termed as the Darcy's friction factor and is related to the Reynold's number and pipe roughness. The applicable and widely used graphs are given in several text books.

For turbulent region. The friction factor value should be read an appropriate curve for a pipe of roughness ε by calculating its ratio with pipe diameter (ε/D).

The log-log plot is difficult to read and the reading is error prone due to non-linearity of scale. Several correlations are therefore proposed by various authors so that the friction factor can be calculated from the Reynold's number. Some of the famous correlations are given later.

In the case of implicit correlations, an iterative approach is necessary to get the value of the friction factor for given value of Reynold's number. Newton-Rhapson method may be used for getting the value in fewer iterations.

Fanning's equation is also used in place of Darcy's equation as follows:

$$\Delta P/\rho = \frac{(4f_F v^2)}{(2gD)}$$

Comparison should show that the Darcy's friction factor is obviously four times the Fanning's friction factor, f_F. While using any friction factor vs Reynold's number graph to read friction factor and then while using it in the formula to calculate the pressure drop, care must be taken to choose the compatible graph and compatible correlation. This is often a source of error.

Another friction factor is also defined by Churchill (which is half of Fanning's friction factor). The corresponding formula for pressure drop calculation thus has a factor 8 in the numerator instead of 4 in Fanning's equation. So, one needs to be really very careful in handling this prevailing multiple definition scenario. Generally, chemical engineering literature uses Fanning's friction factor and Process industry follows the Darcy's friction factor.

If one uses the f vs R_e plot, it is necessary to note whether it is for Fanning, Darcy or Churchill friction factor. There is a simple way to do it which any engineer should know. If you don't, ponder over it a little and you would get it.

Several simplified correlations are available to calculate friction factors from Reynold's number under different conditions of flow. Some of the commonly used ones are given below with reference to the Darcy's definition of friction factor. Suitable multiplying factors must be used to

velo. = $\frac{\text{vol. flow rate}}{\text{Area}}$ → q ; ρ → density

mass flow rate

convert these correlations for other friction factors.

LAMINAR REGION

$$f=64/R_e$$

TURBULENT REGION

Rough commercial pipes, R_e less than 50000:

$$f=56.8\times10^{-10}R_e^{-2}$$

Smooth Pipe, R_e less than 3400000

$$f=19.656\ln\left[\frac{(1.126R_e)}{f^{-\frac{1}{2}}}\right]$$

Blazius equation, fully developed turbulent :

$$f=0.3164R_e^{-0.25}$$

Another Blazius equation :

$$f=0.046\,R_e^{-0.2}$$

Smooth or rough pipe, R_e less than 3400000, developing turbulent flow:

$$f=19.656\left[\frac{1}{0.27\frac{\varepsilon}{D}+\frac{0.888}{R_e f^{-1/2}}}\right]$$

Most f vs R_e plots would mark transition between developing turbulent flows by a broken line. Most flow situations in process in industry would fall in the fully developed turbulent region and Blazius equation (especially the one with R_e with exponent –0.2) given above is widely used.

The roughness factor ε is dependent on the pipe material and method of fabrication and some representative values are given in the Table 1. Note the wide variation in perceptions of the roughness by different authors. In most plots, Moody's roughness values are used. Because of the variation in friction factor definition and roughness values, it is advisable to stick to one plot with full knowledge of the friction factor it pertains to and the roughness values it refers to.

The frictional pressure drop calculated by any of the above methods should be multiplied by the effective length of the pipe segment to get the net frictional drop across the segment. This is then used in the Bernoulli's equation to obtain the actual pressure drop between pipe origin and destination. The effective length is the actual pipe length if the pipe line is straight and long enough so that pressure drop due to extra turbulence created at the entrance when fluid enters the pipe from an equipment or at the exit when the pipe feeds into another equipment are relatively insignificant as compared to overall frictional pressure drop. In case the pipe has fittings such as elbows, tees, valves, expanders, reducers, etc., an hypothetical straight pipe length of same diameter as the run pipe on which the fittings exits is added in place of each of the fittings. The effective length is the sum of the straight-run pipe length plus the total equivalent for all fittings. Entrance and exit of fluid in and from the pipe segment also adds to turbulence and to extra pressure drop. This effect is also incorporated by adding equivalent length of these. The actual equivalent lengths for important fittings are

given in real terms (i.e. length of pipe to be added) in Tables 2-5. (The tables are taken from the famous paper on practical pressure drop calculations by Robert Kern)

In another approach, equivalent length of fittings are mentioned in terms of diameters of the pipe. This number should then be multiplied by the pipe size to get the equivalent length of pipe to be added. The equivalent lengths for valves and fittings in terms of diameters are reported in several books and are not given here. Analysis of the actual equivalent length for fittings of different sizes as given in Tables 2-5 should show that the equivalent diameter approach is rather approximate. Using actual pipe length as per tables is a more accurate approach.

Above procedure is applicable to fluids, i.e., liquids and gases.

In cases the temperature varies across the pipe segment, the physical properties vary. Also if the fluid is gas/vapor, its volumetric flow rate may vary due to pressure changes arising out of temperature change as well as due to pressure drop. To account for these effects, it may be a good practice to divide the whole line into segments over each of which, the temperature change is not so significant as to change the properties drastically. The properties are suitably updated to incorporate temperature and pressure changes as one traverses these hypothetical segments. Calculation over all the segments thus gives the total pressure drop.

Change in pressure across the pipe may be of importance in case of compressible fluids. It may be ignored if it is less than 10% of the total fluid pressure. However, if it is more than this engineering tolerance, above approach of segmenting the pipe line may be adopted.

A good practice would be to calculate pressure drop over the pipe run assuming fluid properties at inlet or average temperature/pressure conditions to begin with. If the pressure drop so calculated is within 10% or less of the actual pressure levels at which the fluid is flowing, one may ignore the effect of temperature/pressure change. If the pressure drop exceeds 10% of flow pressure, the above approach of segmenting may be restored to.

TWO PHASE PRESSURE DROP CALCULATIONS

Pressure drop in the case of a two phase flow is dependent on the flow regime. For two phase flow conditions, 7 regimes are possible as shown in Fig.3. Flow regime identification is done by following Baker's procedure.

Two Baker parameters B_X and B_Y are calculated as follows:

$$B_Y = 2.16 \frac{W_v}{\left[A(\rho_l \rho_v)^{0.5}\right]}$$

$$B_X = 531(W_l/W_v)\left[\frac{[(\rho_l \rho_v)^{0.5}]}{\rho_l 2/3}\right]\left[\frac{\mu_l^{1/3}}{\sigma_l}\right]$$

In the above definitions, following units are used:

W_v - Vapor flow rate, lb/hr

W_l - Liquid flow rate, lb/hr

ρ_v - Vapor density, lb/ft^3

ρ_l - Liquid density, lb/ft^3

A - Internal cross-sectional area, ft^2

μ_l - Viscosity of liquid, cp

σ_l - Surface tension of liquid, dyne/cm

Note that although the Baker parameters are dimensionless, the numerical constants (2.16, 531) in above equations are dimensionless. Given units must be followed.

The Baker parameter values are than used to identify the flow regime from the plot given (Fig.4). Remember, slug flow must be avoided in process piping applications.

The pressure drop calculations then proceeds as per several correlations offered by several researchers. Only two commonly used ones discussed here.

LOCKHART MARTINELLI METHOD

Assuming that that only the liquid flows in the pipe line, calculate the pressure drop that it would cause over unit length, $(\Delta P)_L$. Similarly, considering that only vapor/gas flows in the pipe, calculate the pressure drop per unit length, $(\Delta P)_V$. Single phase correlations are to be used in getting these two pressure drops.

Lockhart Martinelli Modulus, X, is then defined as follows;

$$X^2 = (\Delta P)_L / (\Delta P)_V$$

For this value of modulus, a multiplier Y_L or Y_V is then read from the plot in Fig.5 and it is appropriately used in one of the following relations to get the two phase pressure drop, $(\Delta P)_{LV}$ per unit length. Multiplying this with the effective length (after including equivalent lengths of the fittings) of the pipe, one gets the total two phase frictional drop.

$$(\Delta P)_{LV} = Y_L \, (\Delta P)_L$$

$$(\Delta P)_{LV} = Y_V \, (\Delta P)_V$$

BAKER'S METHOD

Depending on the regime identified earlier, an appropriate correlation or plot is used to get Baker's modulus, φ and it is multiplied with pressure drop with only gas flowing to get the two phase pressure drop. Fig.6 is used for dispersed flow.

$$(\Delta P)_{LV} = \varphi^2 (\Delta P)_V$$

These correlations were derived by the respective authors by extensive experimentation on air-water flow, but mostly on smaller diameter pipes. There applicability for larger dimension industrial pipes is suspect. However, these remain the most used correlations. Better approaches to two phase flow pressure drop estimation are avaliable but are seldom used.

In two phase flow calculations, confidence levels are low. Also, it is not safe to overdesign here as the flow regime may change and one may get an undesirable flow regime such as slug flow. Extreme precaution is therefore necessary at engineering stage in designing pipes for two phase flow and must be ready to handle problems that may surface at the commissioning stage.

The Baker map is applicable only if the flow line is horizontal. Inclination has a great effect on flow pattern and the flow regime may change for same vapor and liquid flows in same size pipe line if the inclinations are different. Also, in inclined pipes, it matters whether the flow is upward or downward. Extensive work has been reported on these aspects but industrial practices ignore this fact.

If 2 immiscible liquids (water, oil) is to be calculated in multiphase using Baker's method, use differnt surface tension of both.

MULTIPHASE PRESSURE DROP CALCULATIONS

Two immiscible or partly miscible liquid phases and a gas phase comprising of vapors of these liquids and/or other gases give rise to three phase flow situations. There are no reported reliable pressure drop calculation approaches for three phase flow. What is proposed here is a possible extension of the Lockhart Martinelli approach which was reasonably successful in using single phase flow correlations and predicting two phase flow pressure drop. The approach would be something like this:

Step 1
Consider only that the liquid phase including the two liquids is flowing through the pipe. Let these liquids be I and L . Using Lockhart Martinelli method or other method (say Baker's), calculate the pressure drop per unit length that would be caused in this case. Let this be ΔP_u

Step 11
Consider only gas/vapor is flowing and calculate the pressure drop that would occur per unit length using single phase pressure drop correlation. Let this be $\Delta P\sigma$

Step111
Calculate the Lockhart Martinelli modulus as was done in the two phase flow situation as follows:

$$X^2 = \Delta P_U / \Delta P_G$$

not self draining [illegible] CV

[illegible]

Step1V
For this value of modulus, a multiplier Y_L (i.e.Y_U) or Y_v (or Y_G)is then read for the plot in Fig. 5 and it is appropriately used in one of the following relations to get the three phase pressure drop, $(\Delta P)_{UV}$ pre unit length (after including equivalent lengths of the fittings) of the pipe, one gets the total three phase frictional pressure drop.

$$(\Delta P)_{UV} = Y_L \Delta P_U$$

$$(\Delta P)_{UV} = Y_G \Delta P_G$$

It may be appreciated that this is nothing but using the Lockhart Martinelli approach on itself. In absence of any other correlation with proven m erit, this is likely to be a good engineering approach.

PIPE SIZING

The earlier mentioned three pipe sizing approaches are discussed here in brief.

PIPE SIZING BASED ON VELOCITY CONSIDERATIONS

This is the simplest of approaches. Herein, recommended values of linear velocities for the flowing medium are used along with the design flow rates to back out the pipe diameter. Recommendations for the linear velocities may arise due to process considerations, material of construction considerations, corrosion considerations, economic considerations based on prior experience etc. or a combination of these. Consider the following examples:

a) In a steam carrying pipe, if the linear steam velocity is beyond a certain value, the flowing steam may pick up the condensate, break it up into fragments. These entrained condensate droplets may

for control valve, [illegible] ΔP ↓

[illegible] CV

impinge against the pipe wall causing erosion and erosion-corrosion.

b) Too low a steam velocity in steam headers may mean a large diameter pipe for design requirement of steam. This would increase pipe cost, insulation cost, etc. thereby adversely affecting economics.
c) A gaseous steam carrying particulates (such as pneumatic solid transport lines) must flow above a minimum velocity to eliminate solids settling down at pipe bottom causing flow obstruction, increased pressure drop etc.
d) A gaseous steam carrying particulates must not flow above a certain linear velocity to eliminate severe erosion of pipeline or elbows etc.
e) A line carrying two phase must be of suitable dimension so that certain two phase flow regimes (such as slug flow) are avoided or a certain regime is guaranteed (such as concentric flow).
f) Linear velocities in exhaust lines should be below certain upper to keep noise within acceptable levels.

These are just representative examples to help appreciate the origin of such restrictions on linear velocities of flowing medium.

Some of the more accepted linear velocities in a variety of design cases are complied in Tables 6 and 7.

PIPE SIZING BASED ON AVAILABLE PRESSURE DROP

This is a more involved method of pipe sizing and perhaps the most important. Pipes are sized here to meet certain process requirements. These process r equirements a re t ranslated i nto the maximum hydraulic pressure drop that one can accept over the pipe segment of interest. A minimum pipe size which causes a pressure drop at the most equal to this maximum acceptable pressure drop is thus recommended. Any size more than this size would also be acceptable, but would be uneconomical as it would involve higher capital cost.

The procedure would be one trial and error. A commercial pipe size would be assumed in terms of NB. The pressure design of the pipe would decide the schedule. From the appropriate tables, the ID of the pipe size would be obtained. Taking this as the hydraulic diameter and for the design flow rates, hydraulic pressure drop over the proposed pipe route is calculated using appropriate pressure drop correlations. If this pressure drop is more than the acceptable level, a higher pipe size is taken for next trial. If the pressure drop is much smaller than that acceptable, next lower pipe size can be tried. Minimum pipe size m eeting t he p ressure d rop r equirement is recommended.

Some important situations where pipe sizing needs to be done using avaliable pressure drop considerations are as follows:

1. Suction Pipe Sizing for a pump: A liquid is to be pumped from a storage tank to an equipment. The storage tank pressure is fixed. On its way from the storage tank to the pump suction, the liquid would loose pressure due to frictional pressure drop. If this pressure drop is excessive, the fluid pressure as it is delivered to the impellers may be below the vapor pressure of the liquid at flowing temperature. The liquid would flash and some of the liquid would then evaporate. As the impellers impart kinetic energy which is then converted to higher fluid pressure inside the pump body, the pressure again rises above the

vapor pressure. The vapor bubbles previously formed thus collapse back into liquid form. This sudden collapse creates the ' cavitation ' effect which could damage the blades and cause vibration and noise. This must be avoided at any cost. It is therefore imperative that pressure drop in the suction pipe should be such that the liquid is delivered to the pump at not less than the vapor pressure at flowing temperature.

2. Even when there is no pump above consideration would apply. During its passage through the pipe, the pressure of the flowing liquid should not drop below its vapor pressure flowing temperature. Otherwise vaporization would take place.

3. In the case of a feed to distillation column, it may be the process requirement that the feed is a saturated liquid. That is, at the flowing temperature, the feed is at vapor pressure and flashes as soon as it enters the column. The pipe carrying the liquid from the reservoir or the previous equipment to the distillation column must ensure that the pressure drop is such as to deliver the liquid at saturation point.

4. A liquid is required to flow at design rate by gravity from a vessel to a lower destination. There is only one pipe size which would come close to this requirement. The nearest commercial size should be recommended

5. A distillation column uses thermosyphon reboiler. This kind of a reboiler works on the principle of natural circulation developed due to a static head difference between the downcomer and riser. Pipe sizing is a delicate balance between barometric head that is avaliable and pressure drop in downcomer and riser.

6. A fluid is to be transported from point A at pressure P1 to point B at P2. There is a flow control valve on the transport line and it has been designed assuming certain pressure drop across the valve is avaliable. Pressure drop across rest of the line that is avaliable is thus limited and pipe must be sized accordingly. This situation can come even in two phase flow lines.

Pipe size as per avaliable pressure drop is
closely linked to process requirements. Any errors in appreciating this and mistakes in pipe sizing could mean that the gravity flow would not sustain, thermosyphon reboiler cannot be commissioned, pump would be damaged and so on.

It helps to appreciate these process related limitations through working out suitable practical cases.

ECONOMIC PIPE SIZING : LEAST ANNUAL COST APPROACH

If the linear velocity and avaliable pressure drop constraints are not stringent or these constraints still leave a scope of a reasonably broad choice of pipe sizes, the most economic among these should be chosen.

The economics is governed by the capital cost of the pipe and accessories including fittings, insulation, etc. and the annual operating cost. If for given service, a smaller size is used, the capital cost would be lower. At the same time, smaller would mean higher fluid pressure drop and therefore higher pumping costs. These two

G = 1-5% of capital cost

F = 7 to 8 times for process industry / 8 to 9 for cross country pipeline

conflicting effects of pipe size mean that there is an optimum pipe size.

For the two costs to be compared, it is necessary that the capital cost be annualized. Fig. 7 shows a typical annualized cost of a pipe for given service as a function of pipe diameter. The operating cost curve is shown in Fig. 8. The sum of these two costs (Fig. 9) gives the total annualized cost which passes through a minimum. The objective of the Least Annual Cost (LAC) approach is to obtain this optimum diameter. Although conceptually simple it is dependent on the reliability of cost data and cost projections over the life of the pipe being designed. A possible approach which appears reasonably scientific and practical is presented here (Nolte, 1978).

The cost of unit length of run pipe of diameter D is calculated as ;

$$C_D = 0.353 \times D^{1.5}$$

(Why?) [will depend on dia.]

cost ∝ $D^{1.5}$, $X = kD^{1.5}$, $k = X/D^{1.5}$

X is the cost of 2 inch diameter pipe of same material and schedule.

The pipe will have certain accessories such as piping elements. Although the cost of these would be application specific, a general process plant average statistics such as the following could be useful to calculate the cost of accessories per unit length as some factor F of the run pipe cost. For example, a typical pipe line (93.5ft) may have 1.6 gate valves, 10.2 bends, 5.9 flanges, 2.1 tees, 32.6welds. So the total capital cost is (1+F) C_D. If the amortization rate is A_M, the annualized capital cost of the pipe and accessories is A_M (1+F) C_D. If the annual maintenance cost is a fraction G of the capital cost, the total pipe cost (capital + maintenance) is (A_M +G)(1+F)C_D. Substituting the expression for C_D in this, one can write the annualized capital plus maintenance cost, C_P as a function of diameter, D, as follows:

factors depends on dia (1/D^1.5)

$$C_P = 0.353(A_M + G)(1+F) \times D^{1.5}$$

The second component is the operating cost involved in pumping the fluid through the pipe. The frictional losses decide the energy lost. If ΔP is the hydraulic pressure drop (say in psi) and W is the fluid flow rate (say lb/hr), the energy expanded in the fluid flow is (W/ρ)(144ΔP). ρ is the density (lb/ ft^3)and the factor 144 in second parenthesis is simple to convert psi into psf for consistency of units.The energy required is then in ft.lb force. The pump has to supply this force using electrical energy. Taking the pump efficiency (E), the annual usage of the pipe in terms of hours of operation per year (Y) and the cost of electrical power, K, (say per KW.hr), the annual energy cost of pumping (C_F) can be written as:

$$C_F = 0.0000542\left(\frac{W\Delta PYK}{E\rho}\right)$$

The units of cost (e.g. Rs. or $ should be same as that power cost). The factor 0.0000542 comes only because of different energy units used for energy (ft.lb.force and kWhr).

The pressure drop, ΔP, can be calculated by conventional methods discussed earlier. One of the simplified forms of pressure drop equations recommended by Generaux has the following form;

For same design condition, 8" - 40 SCH & 16" - 40 SCH, ... table thickness ..

$$\Delta P = 0.1325 \frac{W^{1.84} \mu^{0.16}}{\rho D^{4.84}}$$

It is a dimensionless equation and the units for various quantities are as follows:

ΔP psi
W 1000 lbs/hr
μ cp
ρ lbs./ft^3
D inch

Substituting this in the earlier equation, the cost of moving the fluid per year is

$$C_F = 2840000 \left(\frac{W^{2.84} \mu^{0.16} YK}{D^{4.84} \rho^2 E} \right)$$

Remember, W above is in 1000 lbs/hr.

The total annual cost of unit pipe length is thus

$$C_T = 0.353(A_M + G)(1+F) \times D^{1.5} +$$

$$2840000 \frac{W^{2.84} \mu^{0.16} YK}{D^{4.84} \rho^2 E}$$

The optimum diameter,which minimizes C_T as obtained by differentiating C_T with respect to zero and simplifying is given as follows:

$$D_{optimum} = \left(\frac{W^{0.479} \mu^{0.027}}{\rho^{0.337}} \right) \left[\frac{0.0657YK}{(A_M + G)(1+F)} \right]^{0.169}$$

Most quantities in the above expression are project specific. Their values themselves may not be very reliable. What is then the sanctity of the optimal value of D arrived at ? Some order of magnitude analysis should resolve this issue and give an idea as to how accurately one should try these project specific parameters.

For example, in the expressions in square bracket of the above expressions, one would have reasonably good idea of Y, K, E, X. However, at the time of pipe sizing which is done quite early in the project life, values of a, b, F etc. may at most be guestimates. The important point to note is that the impact of error in estimating the expressions in the bracket is diluted to a great extent by the exponent 0.169. For example, a 33% error in the value of the bracket expressions would lead only to a 8% error in the optimal size estimate. Another parameter which is often a source of low confidence level is the viscosity. But, due to a small exponent of μ in the expressions, one can verify that even a 10 fold increase in viscosity changes the optimal diameter by only 6%.

In view of the above, the optimal diameter expression has been further simplified by using representative values for a (0.143. i.e. 1/7), b (0.01), F (6.75), E (0.55), X (1.32 \$/ft), Y (7880 hrs/year). K (0.0218 \$/kWhr) to obtain the following simplified expressions for LAC diameter.

$$D_{LAC} = 1.717 Q^{0.479} S^{0.142} \mu^{0.027}$$

With D_{LAC} in cm, volumetric flow rate Q in m^3/hr ,
S as specific gravity of fluid at 4 centigrade, and μ in kg/cm.sec.

An alternative expressions is as follows.

$$D_{LAC} = 0.276\, Q^{0.479} S^{0.142} \mu^{0.027}$$

With D_{LAC} in inches, Q in US gals/min and μ in cp.

If the estimates of a, b, F, E, X, Y, K for a project are different than the values used in arriving at the above simplified expressions, correction factors can be suitably used. For example if the actual number of hours of operation is Y and not 7880, the calculated LAC diameter should be multiplied by a factor F_D defined as

$$F_D = 0.2196 Y^{0.169}$$

Similarly, if the amortization rate is 'a' and not 1/7, the correction factor should be

$$F_D = 0.728/(a + 0.01)^{0.169}$$

The reader should ponder a little to see how these correction factors are arrived at.

A better idea would be to use the values realistic estimates of the parameters (a, b, F, E, X, Y, K) whenever they are available and use default values given earlier in the absence of such estimates and uses the expressions for $D_{optimum}$ in its unsimplified form.

The values thus calculated may not conform to the commercial sizes. The following procedure is recommended to arrive at the commercial size.

The adjacent commercial sizes on either side of the LAC diameter are identified. Let these be D_L and D_H on lower and higher sides respectively. An hypothetical size, called crossover diameter is then defined as:

$$D_C = D_L^{0.6} D_H^{0.4}$$

If the LAC diameter calculated earlier is above D_C, D_H is recommended. If it is below D_C, D_L is recommended.

A good question to ask would be why exponent of D_C is 0.6 and that of D_L is 0.4 and why not the other way. Why not equal exponents?

With better computing facilities, one may not be required to use the simplified forms of Fanning equations and other simplifications used in the above approach should be justified by availability of more reliable cost data and values of other project specific parameters. The essence of the approach would remain the same.

RECOMMENDED PIPE SIZE

Whatever the approach used to arrive at the pipe size, it must be kept in mind that the pipe sizing activity is being carried out rather prematurely. The actual pressure drops are going to be decided by the actual layout of a particular point-to-point pipe routing. That evolves at a much later stage. Also, over the normal operating life of the plant, the pipes are subjected to modifications in their ID (due to fouling) and surface roughness (due to scaling, erosion, corrosion etc.). Also, optimization exercises and capacity enhancements in future may require the same pipe to carry larger amounts of process fluid. In view of all these, it is an industrial practice to recommend a pipe of one size higher than what is arrived at by any of the above procedures.

(This paper has relied heavily on the article by Robert Kern, published in Chem. Engg. World)

NPSH → absolute pressure

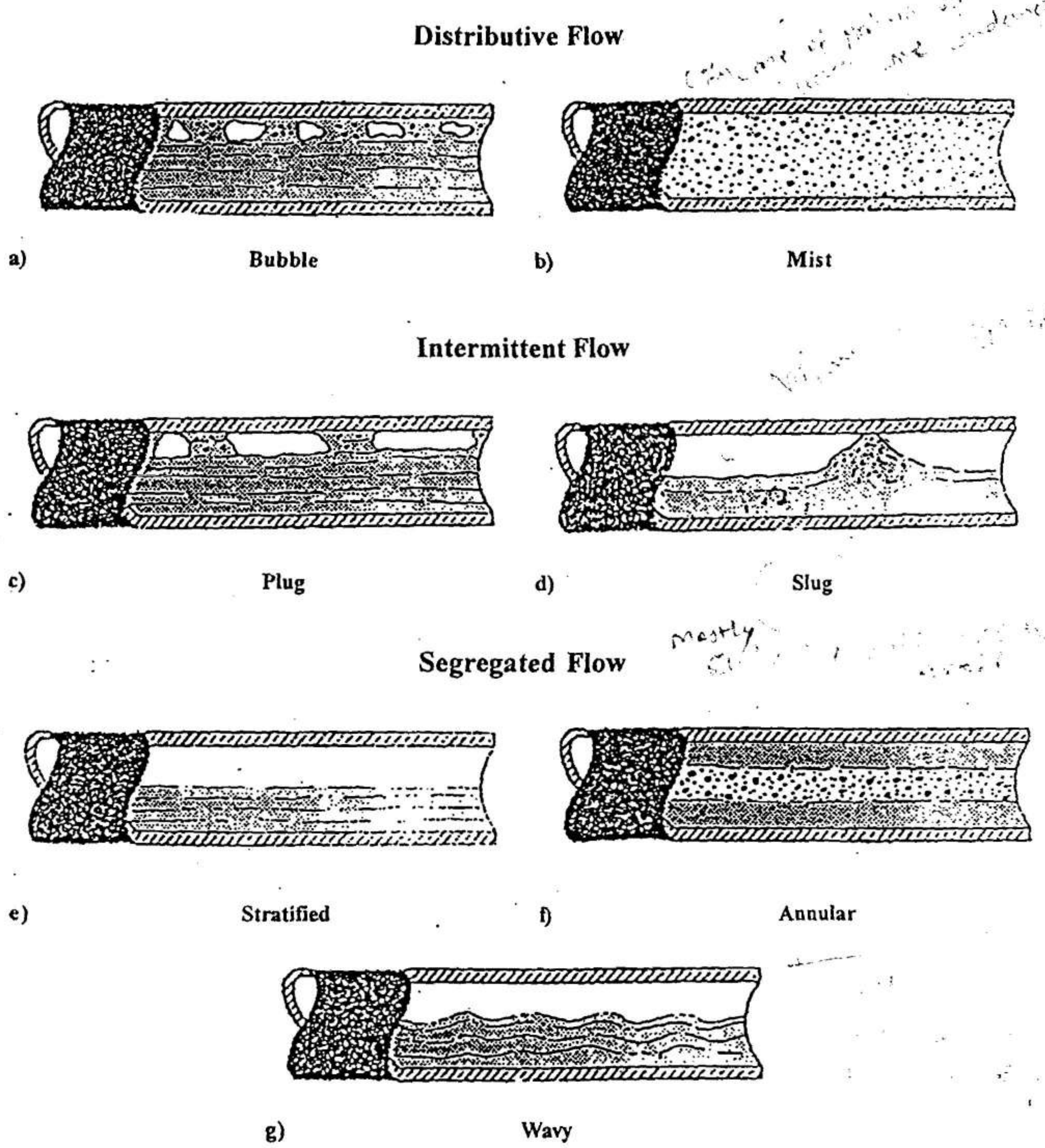
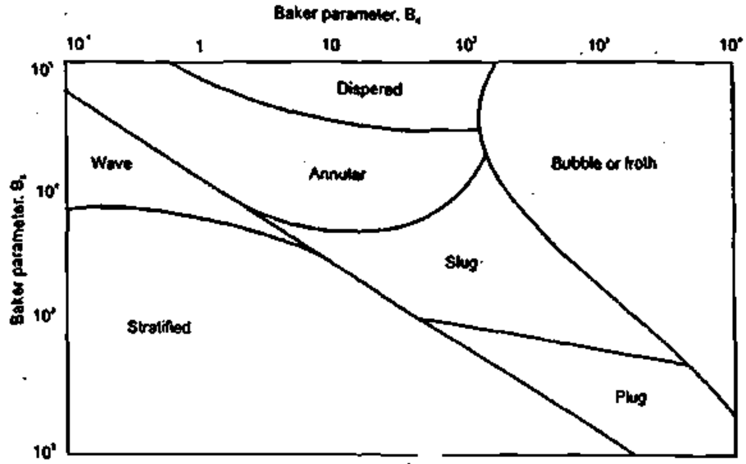


Fig. 3
Flow Regimes for Two Phase Flow

– Slug
– Any
vapour liquid



4

Two-Phase Flow Correlations						
Dispersed	Bubble	Slug	Stratified	Wave	Plug	Annular
Moody or Fannings friction factor dia.	$Ø = \frac{14.2\,X^{0.75}}{(W_L/A)^{0.1}}$	$Ø = \frac{1.190\,X^{0.815}}{(W_L/A)^{0.5}}$ Avoid slug flow	$Ø = \frac{15{,}400\,X}{(W_L/A)^{0.8}}$ Horizontal pipe	Use Fig. 5 and Eq.(9) and (10) Horizontal pipe	$Ø = \frac{27.315\,X^{0.855}}{(W_L/A)^{0.17}}$	$Ø = aX^b$ $a = 4.8 - 0.3125\,d$ $b = 0.343 - 0.021\,d$ d = I.D. of pipe, in For pipe 12-in and over, use $d = 10$.

Fig. 4

Baker parameters determined the type of two phase flow and appropriate two phase flow correlation sets unit loss

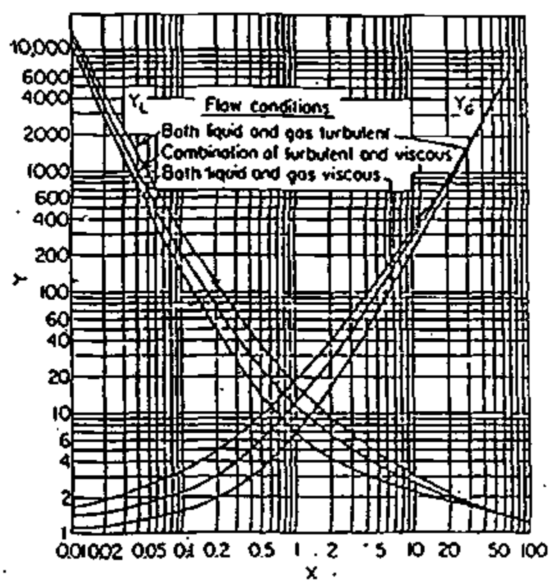


Fig. 5

Parameters for pressure drop in liquid - gas flow through horizontal pipes

[Based on Lockhart and Martinelli, Chem. Engg. Prog., 45, 39 (19.19)]

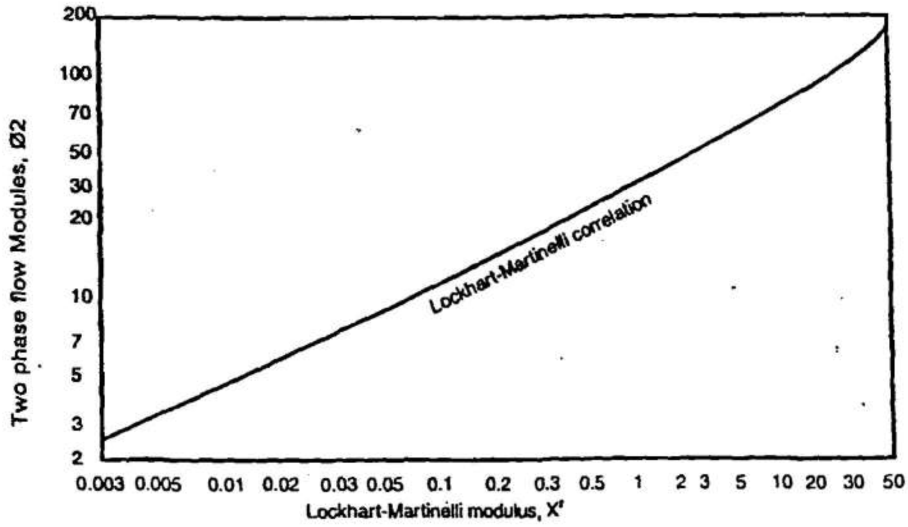


Fig. 6
Lockhart Martinelli Correlation relates vapour and liquid properties to established two phase flow modules.

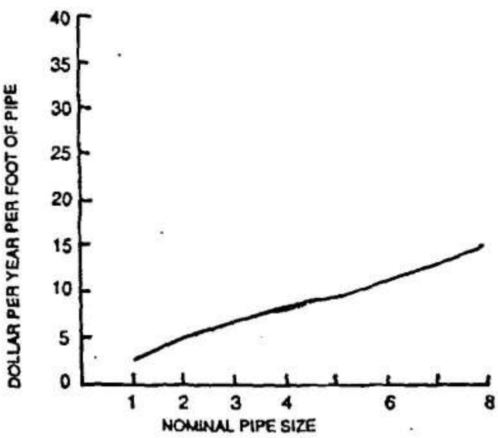


Fig. 7
Amortized capital costs for one foot of pipe.

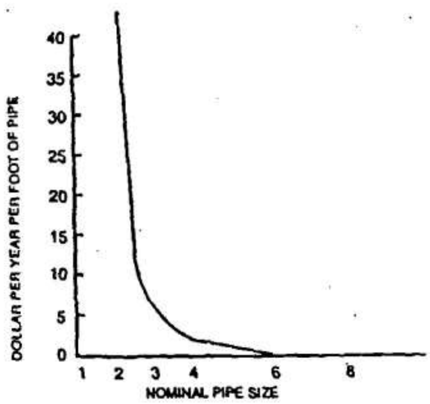


Fig. 8
Annual cost of operating one foot of pipe.

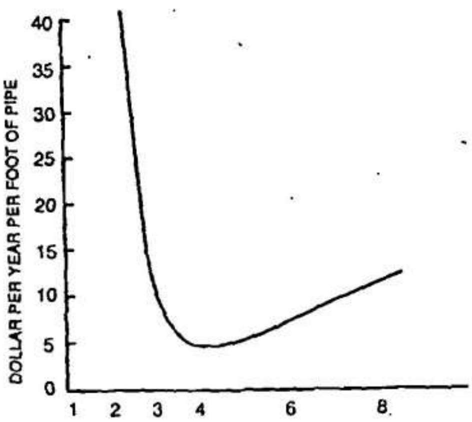


Fig. 9
Total Annual Costs of One Foot of Pipe

Table 1

Surface roughness of various pipes and tubes	Miller (32)		Moody (34)		Kutateladze (29)	
	in.	mm	in.	mm	in.	mm
Tubing						
Drawn	0.0001	0.0025	0.00006	0.0015		
Clean, seamless					0.00006-0.0004	0.0015-0.01
Glass	0.0001	0.0025			0.00006-0.0004	0.0015-0.01
Steel						
New	0.001	0.025	0.0018	0.046	0.0025	0.06
Light rust	0.01	0.25				
Deaerated saturated Steam					0.008	0.2
Condensate (heavy rust)					0.035	0.9
Concrete						
Smooth	0.001	0.025	0.012	0.3	0.03	0.8
Precast	0.01	0.25				
Rough	0.02	0.5	0.12	3.0	0.35	9.0
Cast Iron						
Uncoated	0.006	0.15	0.01	0.25	0.012	0.3
Coated	0.006	0.15	0.005	0.12		
Wood						
Birch veneer					0.001-0.002	0.025-0.05
Pine veneer					0.004	0.1
Rough			0.036	0.9		
Galvanized						
Smooth finish	0.001	0.025				
Normal finish	0.006	0.15	0.006	0.15		

Resistance to Flow for Various Types of Valves - Table 2
(Resistance in equivalent pipe length, ft)

Nominal Pipe Size In	Gate, Fully Open	Globe,[1] Fully Open Bevel or Plug Seat			Check		Straight Through Cock[1]	Three-Way Cock[1]		Butterfly, Fully Open
		90°	60°	45°	Swing	Ball		Straight Through Flows	Flow Through Branch	
1 ½	1.75	46	23	18	17	20	2.5	6	20	6
2	2.25	60	30	24	22	25	3.5	7.5	24	8
2 ½	2.75	70	38	30	27	30	4	9	30	10
3	3.5	90	45	38	35	38	5	12	36	12
4	4.5	120	60	48	45	50	6.5	15	48	15
6	6.5	175	86	72	65	75	10	22	70	23
8	9	230	120	95	90	100	13	30	95	27
10	12	280	150	130	120	130	16	38	120	35
12	14	320	170	145	140	150	19			40
14	15	380	190	160	150	170	20			45
16	17	420	220	180	170	190	22			50
18	18	480	250	205	180	210	24			58
20	20	530	290	240	200	240	27			64
24	32	630	330	270	250	280	33			78

For partially globe valves, multiply tabulated values by 3 for three-quarters open, by one-half open, and by 12 for one-half open, and by 70 for one-quarter open

1. With port area open. Port area = pipe size
2. Port area equals 80% of the pipe area

Resistances of Elbows, Tees and Bends - Table 3
(Resistance in equivalent pipe length, ft)

Nominal Pipe Size in	90° Elbows* Short Radius R=1D	90° Elbows* Long Radius R=1.5D	90° Bends* R=5D	90° Bends* R=10D	Tee Flow Through Branch	Tee Flow Through [illegible]
1 ½	4.5	3	2.5	4	8	3
2	5.25	3.5	3	5	11	3.5
2½	6	4	3.5	6	13	4
3	7.5	5	4	7.5	16	5
4	10.5	7	5.5	10	20	7
6	15	10	8.5	15	30	10
8	21	14	11	20	40	14
10	24	16	14	25	50	16
12	32	21	16	30	60	21
14	33	22	19	33	65	22
16	39	26	21	38	75	26
18	44	29	24	42	86	29
20	48	32	27	50	100	32
24	57	38	32	60	120	38

* For 45° elbows and bends, estimate 50 % of tabulated values

For 180° returns double the tabulated values.

Resistances of Horizontal and Vertical Inlets and Outlets - Table 4
(Resistance in equivalent pipe length, ft)

Resistance Coefficient	K = 1.0	K = 7.8	K = 0.5	K = 0.2
Nominal Pipe Size In				
½	2	1.5	1	0.5
¾	3	2.5	1.5	0.75
1	4	3	2	1
1 ½	7	5.5	3.5	1.75
2	9	7	4.5	2.25
3	15	12	7.5	3.75
4	20	16	10	5
6	36	29	18	9
8	48	38	24	12
10	62	49	31	15
12	78	60	39	19
14	88	70	44	22
16	100	78	50	25
18	120	95	60	30
20	136	107	68	34
24	170	135	85	42

[illegible] 90° Elbow length [illegible] 48"

[illegible]

So, Factor 19 [illegible]

48" [illegible]

liquid < 3 m/s (Three)
Vapour < 50 m/s

Resistance of Eccentric and Concentric Reducers, And of Sudden Changes in Line Size - Table 5
(Resistance in equivalent pipe length, ft.)

Nominal Sizes In. d_1	d_2		
3/4	½	0.6	0.5
1	½	1.2	0.7
	¾	0.6	0.6
1½	¾	1.6	1.0
	1	1.2	0.9
2	1	2.2	1.3
	1½	1.3	1.3
3	1½	3.8	2.4
	2	2.7	2.3
4	2	5	3.2
	3	3	3
6	3	8	5
	4	4	4
8	4	12	7
	6	7	7
10	4	15	8
	6	14	9.5
	8	6	6
12	6	19	12
	8	14	12
	10	6.5	6.5
14	6	22	14
	8	22	14
	10	15	13
	12	6	6
16	8	27	17
	10	23	17
	12	15	15
	14	7	7
18	10	30	19
	12	23	19
	14	15	15
	16	4	4
20	12	30	23
	14	21	23
	16	13	13
	18	5	5
24	18	25	25
	20	12	12

Note : Add these equivalent length to the equivalent length of the smaller pipe and its components.

Typical Liquid Velocities in Steel Pipelines - Table 6
(Resistance in equivalent pipe length, ft.)

Nominal Pipe Sizes in.	2 or less	3 to 10	10 to 20
Liquid & Line	Velocity Ft/S	Velocity Ft/S	Velocity Ft/S
Water			
Pump suction	1 to 2	2 to 4	3 to 6
Pump discharge (long)	2 to 3	3 to 5	4 to 7
Discharge heads (short)	4 to 9	5 to 12	8 to 14
Boiler feed	4 to 9	5 to 12	8 to 14
Drains	3 to 4	3 to 5	-
Sloped sewer	-	3 to 5	4 to 7
Hydrocarbon liquids (Normal viscosities)			
Pump suction	1.5 to 2.2	2 to 4	3 to 6
Discharge header (long)	2.5 to 3.5	3 to 5	4 to 7
Boiler feed	4 to 9	5 to 12	6 to 15
Drains	3 to 4	3 to 5	-
Viscous oils			
Pump suction			
Medium Viscosity	-	1.5 to 3	2.5 to 5
Tar and fuel oils	-	0.4 to 0.75	0.5 to 1
Discharge (Short)	-	3 to 5	4 to 6
Drains	1	1.5 to 3	-

Typical Velocities in Gas and Vapor Lines - Table 7

Nominal Pipe Size In.	Saturated Steam or Saturated Vapor	Super heated Steam, Super heated Vapor or Gas	
	Low Pressure	Medium Pressure	High Pressure
	Velocity Ft/S	Velocity Ft/S	Velocity Ft/S
2 or less	45 to 100	40 to 80	30 to 60
3 to 4	50 to 110	45 to 90	35 to 70
6	60 to 120	50 to 120	45 to 90
8 to 10	65 to 125	80 to 160	65 to 125
12 to 14	70 to 130	100 to 190	80 to 145
16 to 18	75 to 135	110 to 210	90 to 160
20	80 to 140	120 to 220	100 to 170

Note : Within the above velocities and line-size ranges, (a) large lines can have higher velocities than smaller ones (b) short lines, and leads from headers can have higher velocities than long lines and headers.

Reboiler down comner (liquid)	3 to 7
Reboiler, riser (liquid and vapor)	35 to 45
Overhead condenser	25 to 100
Two-phase flow	35 to 75
Compressor suction	75 to 200
Compressor discharge	100 to 250
Inlet, steam turbine	120 to 320
Inlet, gas turbine	150 to 350
Relief valve discharge	0.5 V_c *
Relief valve entry point at silencer	V_c *

* V_c is the sonic or critical velocity.

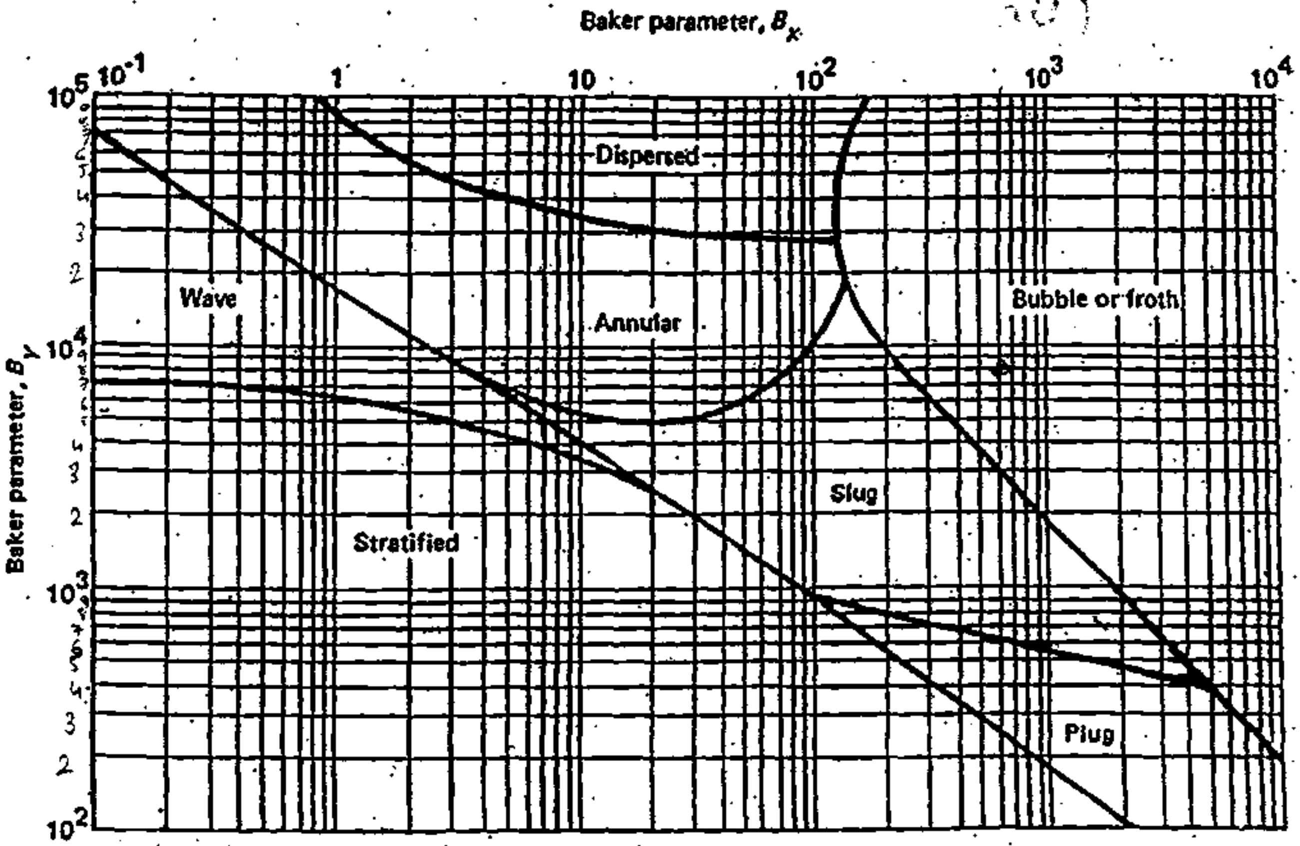
$$\frac{LD_1}{LD_2} = \left(\frac{D_1}{D_2}\right)^{4.8}$$

(upto Re = 1,00,000)

after that

$$\frac{LD_1}{LD_2} = \left(\frac{D_1}{D_2}\right)^{5}$$

For horizontal pipe. But generally ...

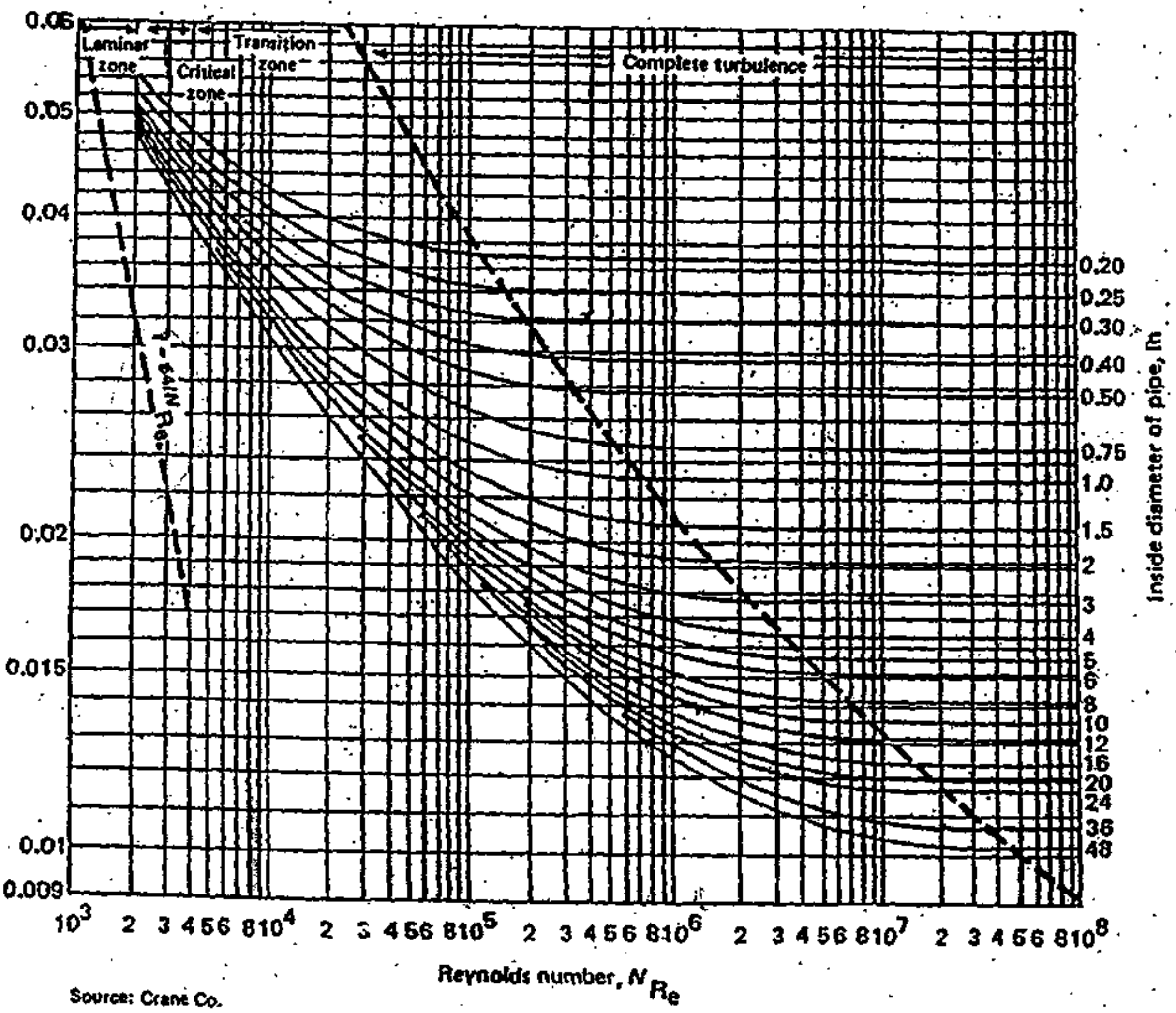


Two-Phase Flow Correlations						
Dispersed	Bubble	Slug	Stratified	Wave	Plug	Annular
Use Fig. 3 and Eq. (3)	$\phi = \frac{14.2 X^{0.75}}{(W_l/A)^{0.1}}$	$\phi = \frac{1{,}190 X^{0.815}}{(W_l/A)^{0.5}}$ Avoid slug flow	$\phi = \frac{15{,}400 X}{(W_l/A)^{0.3}}$ Horizontal pipe.	Use Fig. 5 and Eq. (9) and (10) Horizontal pipe	$\phi = \frac{27{,}315 X^{0.855}}{(W_l/A)^{0.17}}$	$\phi = aX^b$ $a = 4.8 - 0.3125\,d$ $b = 0.343 - 0.021\,d$ d = I.D. of pipe, in For pipe 12-in and over, use d = 10.

Courtesy: Mr. Ovid Baker and The Oil and Gas Journal.

FIG. 1. Baker parameters determine the type of two-phase flow and the appropriate two-phase-flow correlation sets unit loss.

$$\text{Schedule} = \frac{1000\,P}{\text{allowable stress}}$$



Source: Crane Co.

FIG. 2. Friction factors in new commercial-steel pipe for vapor-phase or liquid-phase flow.

$f = 64/N_{Re}$ → Darcy

$f = 16/N_{Re}$ →

$f = \frac{8}{N_{Re}}$ → Churchill

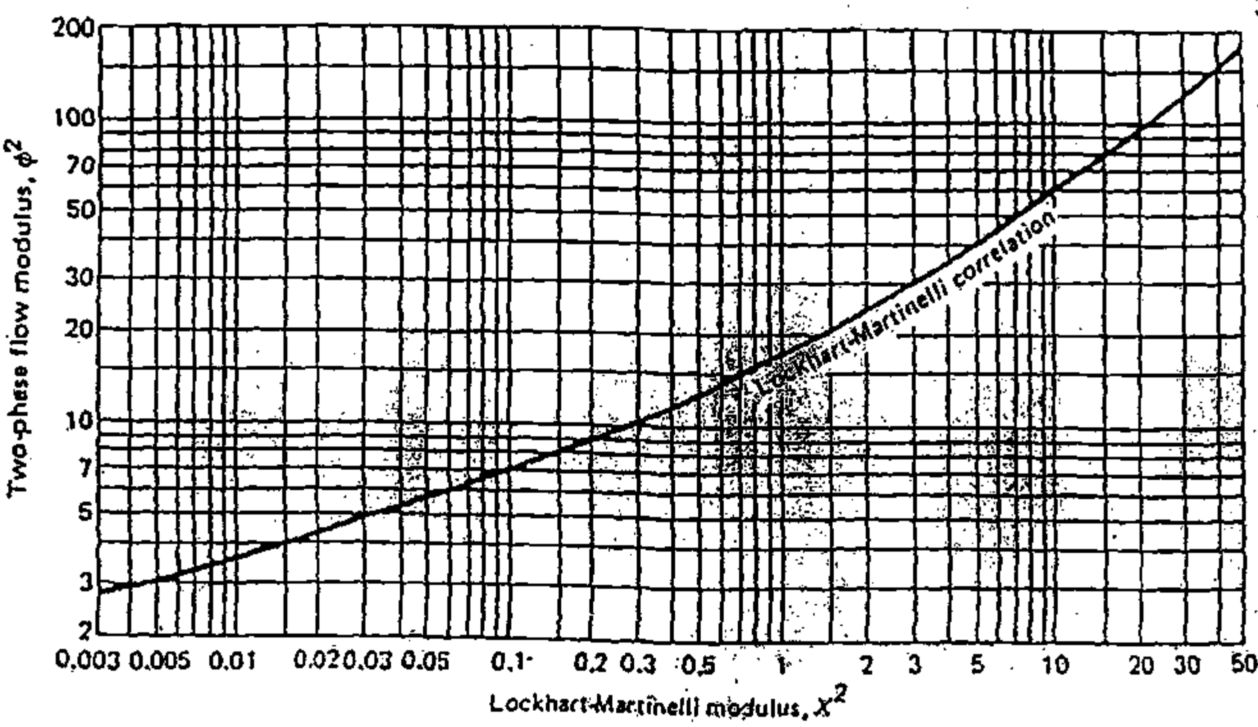


FIG. 3. Lockhart-Martinelli correlation relates vapor and liquid properties to establish two-phase flow modulus.

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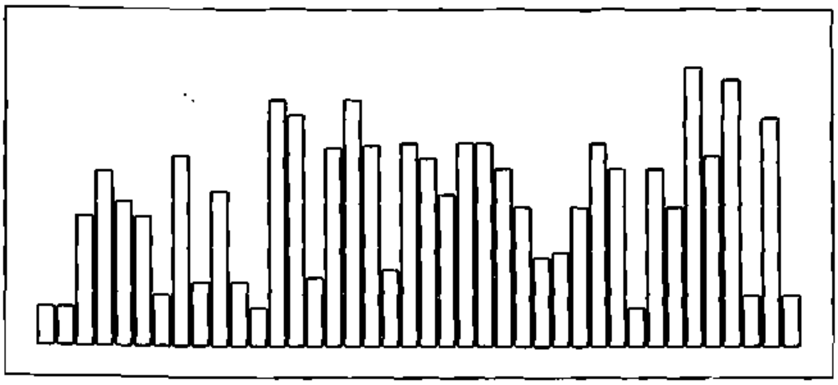

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MECHANICAL DESIGN FUNDAMENTALS

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INTRODUCTION

The life cycle of any chemical process involves inputs from almost all disciplines of engineering and science. The two main contributors are chemical engineering and mechanical engineering. It is very difficult to say who should get more credit for making a chemical process plant function, a chemical engineer or a mechanical engineer. It depends on which side of the great divide you belong to. But the fact is that the success of any chemical process technology is often given to a chemical engineer species for his great process invention and process design. The mechanical engineer who converts his dream process or process dream into reality is often forgotten if the process is successful. Any mishap, however, brings him into focus and the enquiry into such unfortunate episodes unequivocally blames him for it. After all, what fails in a commercial scale plant and causes a shut down or even accident in a process plant is a small or big mechanical component of the plant. Chemical reaction does not fail to occur nor does the vapor liquid equilibrium alter its course. The partisan treatment the two contributor species get is all too noticeable but unavoidable.

One can look at any chemical process plant in two ways. One look, the outwardly look, is at the structures such as vessels (reactors, boilers, columns, tanks, etc.), pipes, pipe components (valves, elbows, expanders, reducers, etc.), machines (pumps, compressors, agitators, turbines, etc.), supports (pipe supports, hangars, buildings, platforms, pipe racks etc.), and facilities (weigh bridge, safety shower, etc.). What catches the eye here is the mechanical and civil aspects of a chemical process plant. Another look is at the outwardly invisible things and happenings. These include the reactions that take place in a reactor, coke or fuel that burns in a boiler, vaporization that takes place in an evaporator or boiler, radiation that heats in a furnace, pressure drop that takes place in a pipe, condensation that takes place in a steam header, crystallization that takes place in a crystallizer, etc. Traditionally, chemical engineers confine their attention to what happens inside the structures and other engineers such as mechanical engineers, civil engineers worry about what happens to the structures.

Most of the operations listed above occur at temperatures and pressures, which are different from normal atmospheric conditions. These operations are often hazardous and do put the surroundings at risk. The job of the mechanical and civil structures is to confine these risky operations within vessels and pipes, act as boundaries between these risky but necessary operations and the outer world. While protecting the outer world from risk, these structures suffer stresses and strains themselves. They have their own limitations dictated by their material of construction, method of design and construction/fabrication, schedule of maintenance and their physical age. Any flaw or shortcoming in any of these aspects would mean that these structures would be unable to do their protector's role perfectly and mishaps would occur.

Mechanical engineering and civil engineering designers have to make sure that the structures would guarantee reasonable safety for a reasonable period of time and not fail in spite of continuous or intermittent harsh conditions faced by their designed structures.

In this paper, we will focus on the important aspects of safe mechanical design of process plant structures, especially equipment and piping.

INFORMATION NECESSARY FOR MECHANICAL DESIGN

Mechanical design of equipment and auxiliaries, which together comprise a process plant, follows the process design stage. At this stage, a PFD (Process Flow Diagram) is available. For each major equipment, information such as its capacity, operating temperature and pressure, chemical composition of its contents during operation etc. are available. Similarly for every pipe connection between vessels or machinery (pumps etc.), the flow rate, temperature, pressure and composition are known. A mechanical engineer has now to take important design decisions, which would ensure that his design of equipment and associated piping would not fail.

Failure of a structural part is said to occur when stresses, strains or a certain function of stresses and/or strains in the structure reach a critical point. Any design has to guard against this perceived failure.

The designer must know two things at this stage.

a. How the stresses and strains in his envisaged structure can be calculated from applied load, and

(Identify load & calculate stress, strain)

b. What is the critical combination of stresses and strains at which failure would occur.

The answer to the first (a) lie in the broad domain of applied mechanics and to the second (b) in the domain of physics of solid. The minimum essentials from these areas required to appreciate various design procedures are covered here in brief.

WHAT IS A FAILURE?

Failure of a structural part can occur by

- excessive elastic deformation,
- excessive non-elastic deformation, or
- fracture

Mechanical design of any structure or its component must guard against 'excessive' deformation under extreme conditions that the structure may face during its operation cycle. The extreme conditions may occur during normal operation, start up or testing etc.

Moderate deformation (elastic or non-elastic) may be beneficial in that they can reshape the structure reversibly or irreversibly so as to redistribute the stresses in a structural part and prevent their rise anywhere in the structure to levels at which failure can occur. Moderate deformations can thus give desirable flexibility to any system.

While deformation may be beneficial in most cases, in some cases, it may lead to change in the shape of the body that causes an increase in the stresses for the same applied load. This increase in stresses may increase the deformation further, which leads to further increase in stresses and so on. This may continue till fracture or rupture takes place. Elastic buckling or plastic unhindered extension

of a rod or wire during the course of which its cross-section diminishes and the stresses for an applied load increase is the simplest example. Specimen tests for determination of material properties such as modulus of elasticity, ultimate tensile stress or strength, fatigue behavior under cyclic load etc. tests the material behavior till such ultimate failure occurs. Some of these properties are crucial design inputs.

SPECIMEN TEST AND IMPORTANT MATERIAL PROPERTIES FOR DESIGN

Mechanical properties of any material of construction are dependent on their chemical composition as well as method of manufacture. The choice of material of construction (MoC) for a given service (fluid to be handled, pressure, temperature etc.) depends on both the chemistry and physics of the MoC.

The chemistry decides the material's interaction with the fluid that it is expected to handle. A suitable material which would not be chemically reactive with the fluid and hence will not be corroded itself and/or contaminate the contents with corrosion products, which is hard enough to withstand the erosive, abrasive action of the fluid/solid that is being handled, etc. is important to choose. Choice of material of construction from this point of view requires expertise and knowledge of metallurgy and chemistry. Broad guidelines for choice of MoC have emerged. These are, however, outside the scope of the present topic.

Another important aspect is that the material so chosen be adequately strong so that any structure, which is made out of it, can withstand process conditions for reasonable amount of time. Some basic concepts from strength of material are involved here and are briefly discussed to prepare the background for design.

Most materials of construction used in the process industry are metallic in nature. Pure metals or alloys of various compositions are used for given service. While the first choice is dictated by the chemistry between the MoC and the process fluid, the ultimate choice from among various options, which satisfy chemistry considerations, is on the physics of the MoC. The important properties used in design are temperature dependent and are experimentally determined using a specimen test. The results of such tests in terms of properties of various Materials of Construction at different temperatures are then compiled as material standards. Each country has its own standards institution and may also follow several international standards.

A schematic of a typical specimen test apparatus is as shown in Fig. 1.

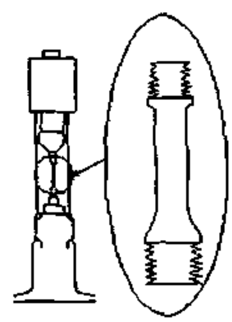


Fig. 1: Typical Specimen Test Apparatus

A specimen piece of a MoC is held firm between the jaws of the test machine and can be subjected to tensile load by pulling the jaws apart with a known force or a compressive load by pushing them closer with known force. The specimen can also be subjected to a cycle of tensile load and compressive load with known amplitude and frequency. Material behavior can be studied under such applied loads and its properties derived.

The two important parameters which quantify the behavior of a specimen are the strain and the stress. For tensile load, the strain is simply the ratio of increase in length of the specimen under constant sustained load to the original length of the specimen before the load is applied. For compressive load, it is similarly the ratio of decrease in length to the original length under sustained load. Strain is thus an observable and measurable quantity as the extension or compression of the specimen can be directly measured. Strain is also a dimensionless quantity.

Stress is defined as the applied load per unit cross-section of the specimen. Unlike strain, it is dimensional. The common units for stress are psi (Pounds per square inch), kPa, MPa, kg. per square cm. etc. depending on the system of units used for force and linear dimension.

When there is no load, there are no stresses and no strain. When a small tensile load is applied, the strain can be measured and stress derived. If the load is removed, the specimen returns to its original shape. That is there is no residual or permanent strain in the specimen. This situation continues up to a certain level of stress. A stress-strain curve in this region is also a straight line (Fig. 2), i.e. stress is proportional to strain.

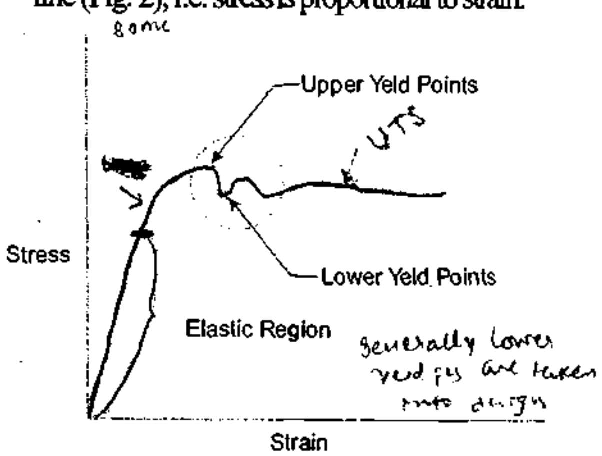


Fig. 2: Typical Stress Strain Curve

Most metals and alloys exhibit this behavior. This region of the curve is called the elastic region, as the MoC's behavior is elastic like a rubber band. As the tensile load during the test is increased further, a situation arises when the specimen does not return to its original dimension even when the load is withdrawn. This is also the load level (or stress level) at which the stress strain curve begins to deviate from the elastic straight-line behavior as shown in Fig. 2. We say that the metal is undergoing plastic deformation in addition to elastic deformation. When the load is withdrawn, elastic deformation is recovered but the plastic deformation stays.

Most metals exhibit an erratic and uncertain stress-strain pattern as the load increases further. The circled region in Fig. 2 shows this. The highest stress that the metal can withstand under sustained load without continuing to elongate under same load is called the upper yield point. There is also a cluster of lower stress values at which there is accelerated strain. The lowest stress values among these are called a lower yield point (Fig. 2).

If the specimen test is carried out with different specimen cut out from, say, the same piece of rod, each may show different location of upper yield point. All such specimen of same MoC would, however, exhibit same lower yield point. The upper yield point depends upon the chemistry of the MoC but also upon the way the molten metal was frozen in a rod or plate mill at manufacturing stage. During this sudden quenching of molten metal, crystals and crystal aggregates get interlocked in awkward position and they get to readjust themselves only during stress tests and when plastic deformation begins beyond the elastic deformation range. This relaxation is reflected in the location of upper yield point. As different portions of the same rod or plate would have cooled down and frozen differently under cold shower during manufacturing, location of upper yield point cannot be said to be a reliable property of MoC but is

obliterated by the manufacturing process as well. The lower yield point is however a material property and depends on chemistry. The lower yield point is what is used as a reliable yield stress value.

Beyond the yield point, the specimen would continue to deform under the same applied load. The strain thus increases. As it happens, the specimen cross-section decreases, same load would then mean higher stresses causing higher strains etc., as discussed earlier. Plastic instability is said to have set in.

Similar tests are carried out at different temperatures. As temperature of test increases, specimen of same material would elongate more for same load as compared to a specimen tested at lower temperature. It would also yield at lower stress value. The material thus becomes softer, so to say, as it is subjected to higher and higher temperatures. This sets the upper limit on the temperature for any engineering material of construction. The stress strain curves at different temperatures for same MoC would look qualitatively as shown in Fig. 3.

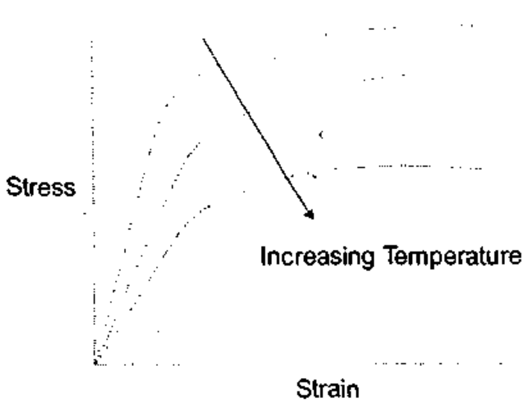


Fig. 3: Stress Strain Curves at Different Temperatures

Stress-strain curves are different for different materials and at different temperatures for the same material. Rather than compiling the curves, specific parameters are derived from the curve and temperature dependent properties of the MoC are reported in material standards. Some of these important properties are as follows.

Modulus of Elasticity:

Also called Young's modulus, it is simply the slope of the straight line representing the elastic line in the stress strain curve. It has the units of stress (as strain is dimensionless). It is an important property and is often used to convert measured strain in a structural element to corresponding stress value if the structural element were to produce that much deformation while still in elastic region. For example, consider Fig. 4.

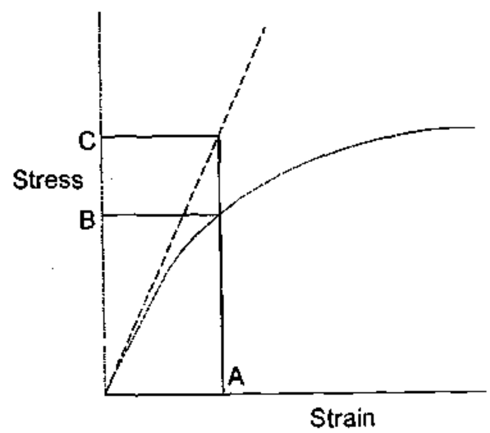


Fig. 4: Code Stresses

Presume that a certain load has caused a strain in an element corresponding to point A on the strain co-ordinate. The actual stress would be corresponding to point B on the stress co-ordinate. However, often in calculation, the stress would be reported as at point C on the stress co-ordinate. Point C is the stress corresponding to the intersection of elastic line with the vertical line corresponding to the strain value at point A.

This method of reporting stresses often causes confusion. For example, how

Code stress = modulus of elasticity x strain

can one have a stress value (corresponding to point C in Fig. 4), which is larger than the yield stress? The structural element should have disintegrated much before reaching that level of stress. The confusion can be avoided if one remembers that this is only a convention followed while doing stress calculations and gives, in a way, an hypothetical stress value. One must remember that stresses, unlike strains, can neither be observed nor measured. They are simply derived quantities. Such back calculated stress values are often called 'code stresses' as they are calculated using strain value and the modulus of elasticity reported in codes.

Yield Stress & Ultimate Tensile Strength:

UTS or Ultimate Tensile Strength or Ultimate Tensile Stress is that value of stress beyond which plastic instability sets in as discussed earlier. Obviously, design should be such that this level of stresses is not reached during the life of any structural element. The lower Yield Point discussed earlier is also called Yield Stress.

Allowable Stress:

Yield Stress is used to decide the allowable stress for any MoC at any temperature by incorporating a suitable safety factor. Allowable stress is often defined as the UTS divided by a safety factor. The safety factor is obviously greater that 1. Designs which ensure that the stress value anywhere in the structure is less than this allowable stress are considered safe designs as they do not allow the structural element to come anywhere close to the point where plastic instability leading to disruption or disintegration of the element would set in. Allowable stress decreases with temperature and an appropriate value applicable for the design temperature should be obtained from the codes. Most often, allowable stress value applicable at design temperature (also called design stress) is directly available from the standards. If not, other available material properties and recommended safety factors should be used to arrive at allowable stress value. For example, if yield stress or 0.2% proof stress value is available at design temperature, the same should be divided by a safety factor of 1.5 to get allowable stress, if yield stress is not available at design temperature but is available at room temperature, the same should be divided by a safety factor of 3.0 to get allowable stress. If stress value for rupture due to static fatigue or creep failure is available at design temperature, a safety factor of 1.5 may be used to get allowable stress. These safety factors are recommended for carbon steel and low alloy steels.

Choice of design temperature is crucial to get the allowable stress. Knowledge of the process is very important here. The knowledge should be not only about the normal operation, but also about start up procedures. How the structure part attains that temperature is also important. Some of the commonly used guidelines are as follows. For parts of a structure, which are not heated directly but attain temperature because they are in contact with the stored or contained material, highest expected temperature of the stored material should be the design temperature. For structural parts, which are heated (say by steam, thermic fluid etc.), highest expected temperature of the heating media or highest expected body part temperature plus 10°C should be the design temperature. Here, 10°C is the safety margin. For fired vessels, parts which are shielded (say by refractory lining), safety margin is 20°C and for

unshielded parts, the safety margin is 50°C. These are only guidelines. What is the highest expected temperature due to unexpected and unintended happenings (such as coolant flow interruption due to control valve closing shut etc.) is for the designer to visualize. What should be the safety margin would depend upon the severity of operation.

Proof Stress:

Also called 0.2% proof stress, It is the stress for 0.2 % strain. In simpler terms, it is the stress value for strain value of 0.002 on the stress strain curve. Two variations of the definition are also in use. In one it is defined as code stress for 0.2% strain. In another, it is the stress value obtained from the intersection with the actual stress strain curve of a line drawn parallel to the elastic line from the point representing 0.2% strain on the strain co-ordinate. See Fig. 5 for explanation of these various definitions. The last definition has been adopted by several international codes.

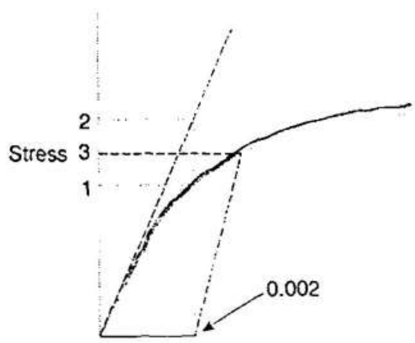


Fig. 5: Proof Stress Definitions

An important point about the proof stress is that it can be altered by preceding plastic deformation or 'cold work'. It can be explained with the help of Fig. 6.

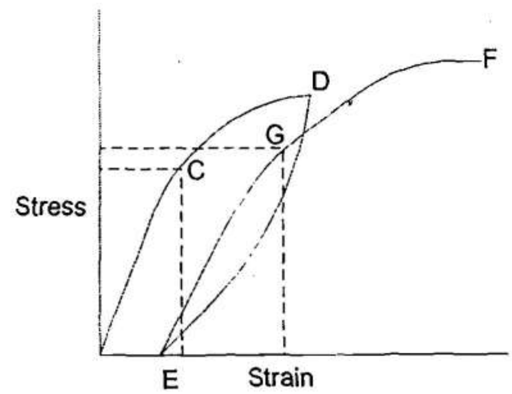


Fig.6: Increase in Proof Stress due to Cold Work

Consider a stress strain curve as shown in Fig. 6. Let a fresh specimen be subjected to gradually increasing tensile loads. A 0.2% proof stress can be marked on curve as the stress value corresponding to point C on the curve. Let the load be increased beyond this point up to point D on the curve. The specimen has surely passed the elastic range and crossed over to plastic deformation. On withdrawal of the load, the specimen would return back to point E with a residual permanent strain as shown. Tensile load test can now be conducted on this specimen, which has seen plastic deformation or 'cold work' previously. The specimen would now follow a stress strain curve with strain zero at point E. Along this stress strain curve (EGF), 0.2 % proof stress is corresponding to point G and is higher than the proof stress for the fresh specimen. The material seems to have hardened with its experience of stress earlier. Most materials show this marginal increase in their proof stress due to cold work.

FATIGUE BEHAVIOR

The above discussed behavior of materials was under sustained loads. The final failure under sustained load due to plastic instability was because the load

carrying cross-section (cross-section of the specimen) diminished to compensate for elongation (as volume of the metal in the specimen remains the same), which led to higher stresses causing further elongation etc. Such failures are termed catastrophic failures. These failures occur almost suddenly as soon as the load crosses a threshold (ultimate tensile strength). These failures take place on the first occurrence of loads in excess of yield stress. Their is another kind of failure, which is not catastrophic in nature but occurs because of the damage done to the grain structure of the specimen due to prolonged application of sustained load and/or tensile-compressive load cycles. The history of applied load plays an important role here and the failure is said to have resulted due to Fatigue.

There are two kinds of fatigues: Static Fatigue and Cyclic Fatigue.

Static fatigue:

Here a specimen fails under a sustained load, which it has withstood for a considerable length of time. The total time for which the load was applied is important here. Whether it was applied continuously or in installments is not important. For example, if a material undergoes static fatigue failure under a sustained load of say X in say 1000 hours, it would do so irrespective of whether

1) the load was applied continuously for 1000 hours, or
2) the load was applied for 10 hours and then withdrawn for 14 hours each day for 100 days, or
3) the load was applied for 5 hours and then withdrawn for 19 hours each day for 200 days, etc.

under a constant load of ~~less~~ same period
M/C fails → static fatigue

The material seems to count only the total time for which it was subjected to the load.

Cyclic Fatigue:

Here a specimen fails under a load cycle, which it has withstood for a considerable number of times. The total number of cycles for which the load was applied is important here. Whether the cycle was frequent or infrequent is not important. For example, if a material undergoes cyclic fatigue failure under a load cycle of amplitude say X in say 1000 cycles, it would do so irrespective of whether

1) the cyclic load was applied once every day for 1000 days, or
2) the cyclic load was applied twice every day for 500 days, or
3) the cyclic load was applied with cycle time of 1 minute and for 1000 minutes etc.

Cyclic fatigue is often expressed as number of cycles to failure for a certain amplitude of load cycle and certain sustained load. For example, a specimen may be subjected to a tensile load increasing from zero to X, then the load gradually withdrawn till it is zero, then compressive load applied gradually till it is X and then it is withdrawn till the load is again zero. This comprises a cycle and it is then repeated again and again. With each cycle, the grains in the material get displaced relative to each other and get more and more interlocked. With each cycle, the material loses its ductility in small increments. A time comes when the grains are so badly interlocked that they cannot allow deformation to withstand load and a small crack develops. This crack grows with further cycles and failure occurs.

Actual fatigue failure of any structural element may be a result of a combination of both these fatigue failure modes.

For example, the above cycles were applied with mean load as zero (the load varied between X and -X). It can also be applied with non-zero mean. For example we may apply a load of Y, then increase it to Y+X, bring it down to Y, lower it further by X to Y-X, increase it to Y, then to Y+X etc. The material in such cases would fail due to static fatigue as well as cyclic fatigue and the cycles to failure would be less as compared to earlier case.

Occurrence of static fatigue failure when the material is under prolonged sustained load coupled with high temperature is also called creep failure.

Major international standards institutions study materials from all such angles and compile data from these extensive/expensive experiments. Most of this data is for solid specimen. These indicators of material behavior obtained from studies on solid specimen are often used to design failsafe vessels such as cylinders and spheres. But these are hollow structures. What is important to note is that failure may occur in such shapes at lower stress values than applicable for solids of same MoC. For example, pressure inside a sealed cylinder can be used as a way of applying load. Pressure, cylinder dimension and wall thickness can be used to calculate stresses in the metal walls. Pressure can be gradually increased till the cylinder bursts. The bursting pressure may be converted to stresses in walls at that pressure. These stress values at failure have been found to be lower than the yield stresses for solid specimen of same material at same temperature. Geometry thus plays a role in failure apart from temperature and chemistry and physics of solids.

A typical sequence of stress values at failure for solid rods and two most important shapes in process industry, namely, sphere and cylinder is rod > sphere > cylinder. This is taken care of by the design formulae for various shapes in some way as will be seen later.

For application at sub-ambient temperatures, more than the above properties, brittleness, hardness etc. become important. These are also quantified and reported by the standards institutes. Their discussion is outside the scope of this paper.

As pointed out earlier, calculation of stresses from applied load is simple for simple shapes such as uniform cross-section solid rods. For more important shapes such as hollow sphere or cylinder, the load is often in terms of pressure exerted by the fluid contained in the vessel and this has to be related to induced stresses in the vessel walls. This is the area of applied mechanics and is the subject matter of the next few sections.

RELATION OF STRESS TO APPLIED LOAD

Process vessels involve several regular shapes. For example, a reactor may have a cylindrical body, an elliptical closure at the top and a conical bottom. A riser in a FCCU (Fluid Catalytic Cracking Unit) may have a larger diameter cylindrical body at the top connected to a smaller diameter cylindrical body at the bottom with a frustum of appropriate cone joining them and a dished (torispherical) closure at top and hemispherical closure at the bottom etc. Pipes transporting process streams from one vessel to another are cylindrical entities. More commonly encountered geometrical shapes in a

process plant are thus: sphere or hemisphere, cylinder, cone or a frustum of a cone, ellipsoid and torus. Design of such body shapes implies decision regarding their wall thickness for given inner or outer overall dimensions which will ensure that under the worst pressure and temperature conditions, these shapes do not develop stresses which would cross the allowable stress limits. For each shape, the codes give simple calculation formulae which are often used without much thought to the roots of such formulae. These formulae are rooted in applied mechanics, safety considerations, fabrication considerations etc. It is proposed to offer here a general applied mechanics approach to the development of relation between stress and applied load, basic formulae for simple shapes to calculate safe thickness for given service and then compare such design formulae to the code procedures for design. This would throw light on how rigorous theory can lead through practical considerations to simple practically applicable design procedures.

For low pressure application

MEMBRANE THEORY OF PRESSURE VESSELS

& For low thickness.

All above shapes have something in common. These are all shapes of revolution. For example, if a semicircular arc revolves through 360^0 around an axis coincident with its diameter, it generates a sphere. A quarter of a circle revolving similarly would generate an hemisphere. A straight line revolving around an axis parallel to it generates a cylinder, a straight line revolving around an axis of rotation at an angle with it would generate a cone or its frustum, a circle revolving around an axis not passing through its centre would generate a torus, a quarter of an ellipse revolving around an axis coinciding with its minor axis would generate an ellipsoidal closure shape and so on. These shapes of revolution are shown in Fig. 7.

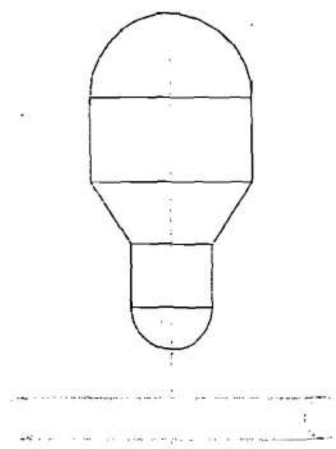


Fig. 7: Shapes of Revolution

If the line that is revolving has a small thickness similar to the thickness of the wall of such shapes, it would generate a practical vessel shape of interest with thickness t. Since the shapes of interest are all shapes of revolution, it would be sufficient if one could establish a way of relating stresses in the wall of an arbitrary shaped vessel generated through revolution of an arbitrary line around an axis of rotation as shown in Fig. 8. We apply concepts from applied mechanics to such a shape subjected from inside to a net fluid pressure P over and above the pressure exerted on the outside of the vessel.

Let us imagine the vessel under consideration as comprised of several tiles glued together as shown in Fig. 8.

(high thickness ½ of diameter)
(lower thickness = 1/6 of diameter)

(according to code of thickness is 1/6 of the dia than this theory may be

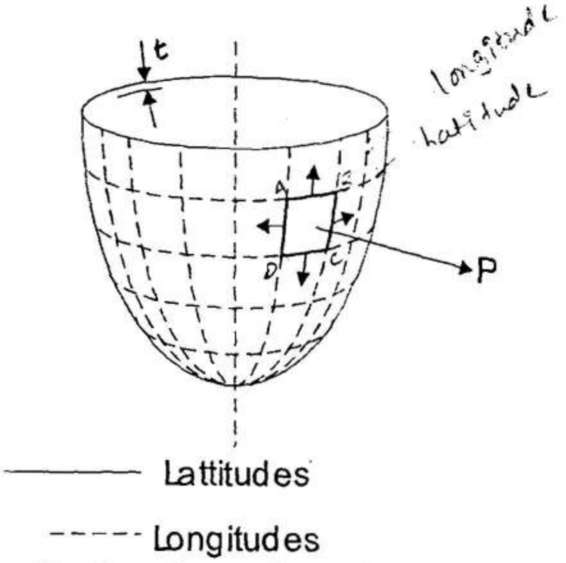


Fig. 8: Arbitrary Shape of Revolution

This mesh of tiles is generated using the longitudes and latitudes of the shape just as we define the longitudes and latitudes of The Earth. The position of the revolving line at several stages of revolution are called longitudes. The loci of different points on the revolving line as the line completes 360^0 of revolution are called latitudes as shown in Fig. 8. Also shown in Fig. 8 is the thickness of the wall of such a vessel which is assumed to be uniform everywhere. We also assume that wall thickness is very small compared to the vessel dimension. Such thin walls are called membranes. The theory to be developed here is called membrane theory.

We consider a tile ABCD out of the whole vessel. It is infinitesimally small in dimension if we choose longitudes, which are separated by a small distance, and also latitudes, which are very close to each other. In that case, even though the tile has a curvature, AB can be said to be equal to CD and AD equal to BC.

The tile is blown up in Fig. 9 with its lateral dimensions as ds_1 (AB or CD) and ds_2 (AD or BC). This tile is being pushed out by the net internal pressure acting on it. Fluid pressure always acts normal to any surface and therefore the net force acting on the tile is a product of the pressure and the area of the tile. If the tile was not glued to its neighboring tiles, it should come out of the vessel wall. It does not because it holds on to other tiles and they hold on to it with appropriate force. These holding forces would have components in the direction of the force due to pressure and these components together exactly nullify the outward push due to pressure. These forces act over the entire wall thickness along ds_1 (AB or CD) and ds_2 (AD or BC). Since the wall thickness has been assumed to be small, the forces can be assumed to be evenly distributed over the thickness. The force divided by the load bearing cross-section gives the stresses. For example the force, which is holding the edge AB with the neighboring tile, is acting on a wall cross-section, which is equal to $ds_1 t$ where t is the thickness of the wall. These stresses are called induced stresses as these are generated because there is a force acting on the tile due to internal pressure. If the pressure vanishes, these stresses also would vanish. The stresses induced in a direction normal to edges AB and CD are denoted as σ_2 and those along edges AD and BC as σ_1 . In view of the small dimension of the tile, the stresses along parallel edges are naturally equal in magnitude and act perpendicular to the edges (see Fig. 9).

meridional/radial [illegible] related to longitudinal [illegible] -

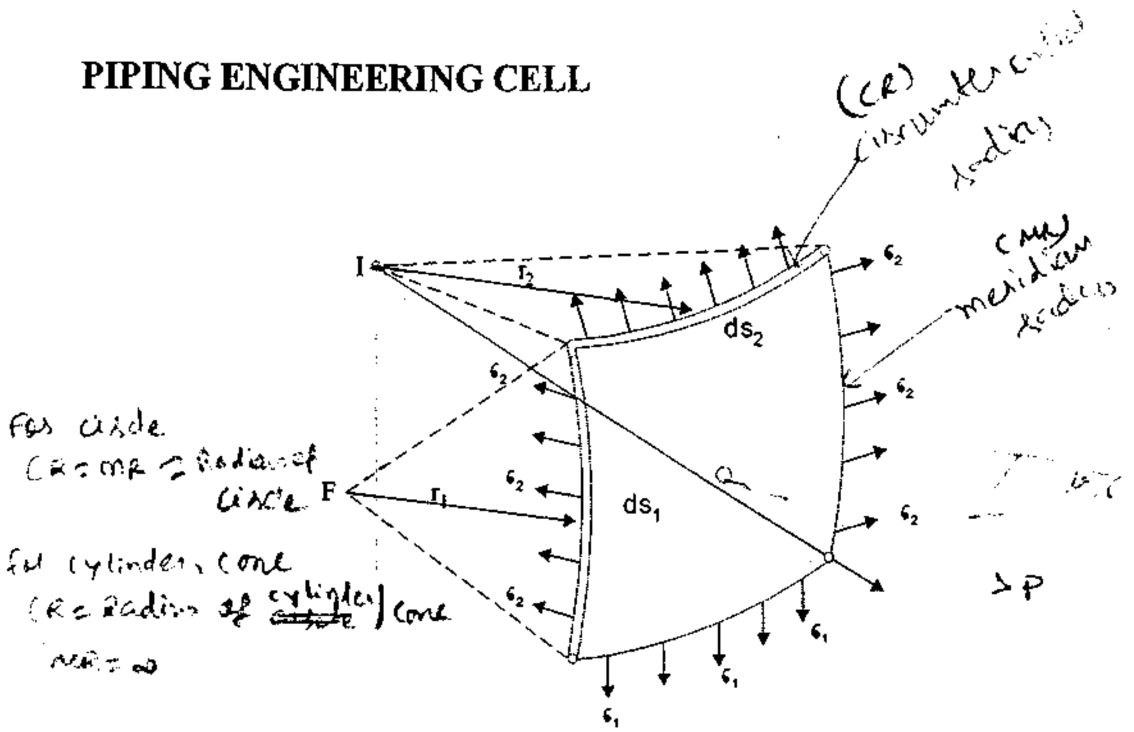


Fig. 9: Differential Element

Let us define two-dimensional parameters of the arbitrary vessel shape. For known vessel shape, these parameters would automatically relate to conventionally used dimensions such as radius of a cylinder etc.

The pressure acts normal to the surface of the tile ABCD. In view of the small dimension of the tile, it can be approximated to act at the center of the tile as shown in Fig. 9. As the tile is a part of the vessel which is symmetrical about the axis of rotation, this direction of the pressure force if extended towards the inside of the vessel would intersect the axis of rotation at some point I as shown in Fig. 9. Also to be noted is the fact that AD and BC are arcs of a circle.

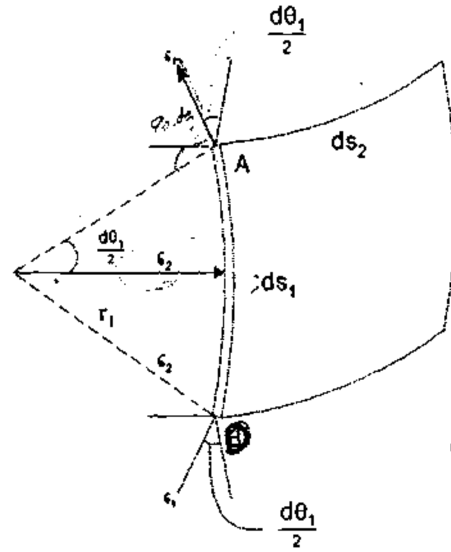
If the circle of which AD (and also BC) is an arc is to be drawn from point I, its radius is called circumferencial radius of curvature. It should be noted that it may not be the same as the conventionally defined radius of a circle where we draw the circle from a point in the same plane as that of the circle. Here, point I may not be in the plane of the circle and the circumferencial radius of curvature may be larger than the conventionally defined radius of a circle.

The radius of curvature of the arc AB (and also CD) is called the meridianal radius of curvature.

Let us now obtain the components of the forces due to induced stresses acting on the edges of the tiles, which oppose the pressure forces.

Consider Fig. 10. Concentrate on the edge AB. Let F be its focal point and r_1 the meridianal radius of curvature. Let the arc AB subtend an angle $d\theta_1$ at the focal point. The stresses induced along edges AD and BC are obviously perpendicular to the lines FA and FB respectively. Let these stresses be denoted by σ_1. These stresses have a component along the direction along which pressure force acts. This direction bisects the angle $d\theta_1$ as shown. The stresses act along edge ds_2, The net force is thus $ds_2.t$ and its component in a direction opposite to the direction of force is $ds_2.t.\ \sin(d\theta_1/2)$. As the situation on edge BD is similar, its component is also

same. The net force from the stresses induced along edges AD and BC is thus $2.\sigma_1.ds_2.t\sin(d\theta_1/2)$.

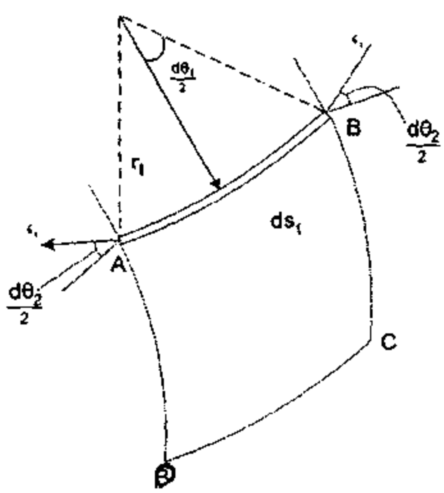


$ds_1 t \times \sigma_2$

Pe = $ds_1 t \times \sigma_2 \cos$
= $ds_1 t \times \sigma_2 \sin \frac{d}{2}$

Fig. 10: Force Components

Similar to the above treatment for stresses along edges AD and BC is the treatment for stresses along edges AB and CD as shown in Fig. 11.



$\sigma_1 = ds_2 t \times \sigma_1$

Fig. 11: Force Components

Concentrate on the edge AD. I is the point of intersection of the pressure force's direction as discussed earlier and r_2 the circumferencial radius of curvature. Let the arc AD subtend an angle $d\theta_2$ at the point I. The stresses induced along edges AB and CD are obviously perpendicular to the lines IA and ID respectively. Let these stresses be denoted by σ_2. These stresses have a component along the direction along which pressure force acts. This direction bisects the angle $d\theta_2$ as shown. The stresses act along edge ds_1 , The net force is thus $ds_1.t$ and its component in a direction opposite to the direction of force is $ds_1.t.\ \sin\ (d\theta_2/2)$. As the situation on edge CD is similar, its component is also same. The net force from the stresses

normal area

induced along edges AB and CD is thus $2\sigma_2 ds_1.t.\sin(d\theta_2/2)$ (S×A = Force)

Force acting on the tile ABCD is $P.ds_1.ds_2$. The tile remains in place because the components of induced stresses exactly balance the outward thrust due to pressure. The force balance is as follows.

$$2.\sigma_1.ds_2.t.\sin(d\theta_1/2) + 2.\sigma_2.ds_1.t.\sin(d\theta_2/2) = P.\ ds_1\ .ds_2$$

The tile ABCD had infinitesimally small dimensions and therefore, from the geometry (Fig. 10,11) following identities hold.

$$ds_1 = r_1\ .\sin d\theta_1 = r_1.d\theta_1$$
$$ds_2 = r_2\ .\sin d\theta_2 = r_2.d\theta_2$$
$$\sin(d\theta_1/2) = d\theta_1/2$$
$$\sin(d\theta_2/2) = d\theta_2/2$$

Substituting these in the force balance and simplifying, one gets the following important result for any arbitrary shape.

$$\frac{P}{t} = \frac{\sigma_1}{r_1} + \frac{\sigma_2}{r_2}$$

This relationship between the circumferential and longitudinal stresses and the geometry of any shape in terms of its two radii (r_1 and r_2) with the wall thickness and pressure can be used for any shape.

The design (that is thickness t) should be such that everywhere in the shape, the stresses should be within the allowable stress limit. It would be necessary for that purpose to get separate relationships for the two stresses σ_1 and σ_2. For a simple shape such as sphere, however, the above relationship is sufficient in itself to find a suitable thickness for a sphere of diameter D.

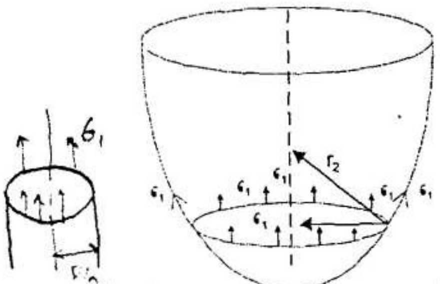
The user should verify that for a sphere, the meridianal and circumferential radii of curvature are equal to the radius of the sphere itself, i.e. D/2. Also, the fact that sphere has a complete three dimensional symmetry provides that the two stresses should be equal, i.e. $\sigma_1 = \sigma_2 = \sigma$. Therefore, for a sphere, one can write;

$$\boxed{\frac{P}{t} = \frac{4\sigma}{D}}$$

Let the allowable stress be Sa. The thickness should be so chosen that σ should not exceed Sa. Or, from the above relationship,

$$\boxed{t \geq \frac{PD}{4\,Sa} \quad \text{or} \quad t \geq \frac{PR}{2\,Sa}}$$

where R is the internal radius of the sphere. For other shapes, it would be necessary to get separate relations for σ_1 and σ_2. This can be done by taking another overall force balance over a section of the shape as shown in Fig. 12.



$t\pi D\sigma_1 = \pi D^2/4\,P \Rightarrow \sigma_1 = PD/4t$

Fig. 12: Force Balance on a Section

Force due to pressure acting on the circular section downwards must be balanced by the force due the vertical component of the longitudinal stress, σ_1, acting on the edge of this circle of width equal to the thickness t. If such a balance is taken, one gets a relation for longitudinal stress as

$$\sigma_1 = \frac{P\,r_2}{2\,t}$$

Substituting this in the earlier derived relation between stresses and pressure, one gets the relationship for circumferential stress σ_2 as follows.

$$\sigma_2 = \frac{P}{t}\left[r_2 - \frac{r_2^2}{2\,r_1}\right]$$

$$\boxed{\sigma_2 = \frac{P}{t}\left[r_2 - \frac{r_2^2}{2r_1}\right]}$$

Reader may check that these definitions of longitudinal and circumferential stresses also lead to same design formula for sphere. For cylindrical shape (generated by a straight line parallel to the axis of revolution revolving around it), the meridianal radius of curvature is infinite (straight line has no curvature) and circumferencial radius of curvature is the same as radius of the cylinder. If this is used in above relationships for σ_1 and σ_2, then one gets the well known result that the longitudinal stresses in a cylinder are half of the circumferencial (Hoope's) stresses. The design should ensure that the Hoope's stress is within the allowable stress limit. A formula for minimum thickness is then obtained as follows.

$$t \geq \frac{P\,D}{2\,Sa} \quad \text{or} \quad t \geq \frac{P\,R}{Sa}$$

where D and R are the diameter and radius of the cylinder respectively. For other shapes, similar approach is to be followed. Getting the two radii of curvature often requires more involved relationships from geometry for these and more complicated shapes.

What is interesting to note is that the design formulae in various codes are similar to these theory-based formulae, but not quite. Comparison of practical design formulae based on internal and external dimensions and the above design formulae is given in Table 1 for sphere and cylinder.

The codes take into account the fact that the shape is going to be made in most cases by welding several pieces together. The portion at and around welded joint may not be as strong as the sheet material elsewhere, unless otherwise tested and proved by techniques such as radiography. The design should be safe even at the weakest portion of the vessel wall. This is taken care of by reducing the allowable stress value by a weld joint efficiency factor, E. The effect is that the calculated thickness would be that much more to take care of the weakness of the weld joint. Another factor in the denominator is pressure dependent. For example, if one considers the formulae based on the internal dimensions, one can clearly see that the effect of this pressure dependent term is to reduce the denominator magnitude as higher and higher pressures are used. This would mean that the code formula would recommend a thickness, which is increasingly higher as the design pressure increases. This makes sense, as the implications of failure of the vessel would be more and more disastrous as the operating pressure increases. This correction in the denominator can be seen to be more for cylinder than for the sphere. This apparently has something to do with the earlier discussed fact that for same material, the spherical shape would fail at stresses lower than the yield stresses for solid specimen and cylinder would fail at even lower stress values. This shape effect is captured in the second term in the denominator of the code design formulae.

Code formulae have been under constant scrutiny. They represent a sum total of theory, safety considerations, fabrication limitations, experience with behavior of designs carried out during the past, acceptance of human limitations in understanding physics of solids and experimentally measuring some of the important material properties and above all, an overriding concern that the design should be done to reduce (to extremely low levels) the probability of disastrous failures causing damage to health of concerned people and environment. Caution is the watchword of mechanical designers, and rightly so.

CONCLUSIONS

The paper attempted to discuss the important basics from physics of solids and applied mechanics, which form the foundations on which practical design procedures rest. The actual designs are subjected to several other sources of stresses and strains such as compressive or bending loads due to own weight, compressive and tensile load due to vibrations in tall structures due to seismicity, wind, eccentricity etc. In additions to the above discussed loads due to internal pressure, which lead to tensile stresses, a structure may encounter compressive stresses due to net pressure acting from outside. This gives rise to the possibility of failure due to buckling and needs to be studied separately. A separate set of design formulae for such special cases is provided by the codes.

Structures, especially pipes also encounter thermal loads. Pipes are installed at ambient temperature. During operation, they carry fluids, which are at super-ambient or sub-ambient temperatures. The pipes would expand or contract. If the pipes are so fastened to equipment or supports etc. that this movement is prevented, compressive or tensile stresses would develop in the pipe wall. Sometimes, pipe sections are forced to change their route. These additional loads on pipe have also to be taken into account while designing and routing a pipe section. Mechanical design of pipes need to consider all such kinds of loads together and still offer safe operation. This necessitates detailed analysis and a broad area of stress analysis has evolved for study of critical pipe sections.

Even for vessels, just designing the body is not sufficient. There have to be openings on the body to provide for several things such as inlet and outlet of process fluids, drainage, inspection, instrumentation, maintenance etc. These openings would weaken the portion surrounding them and need to be specially considered for possible reinforcement. The vessels have to have closures, which need to be fastened in a leak proof manner to the vessel body. The design of flanges, gasket, nuts and bolts is a part of vessel design. For pipe-to-pipe connection or pipe to equipment connection, similar design of flange, gasket, bolts etc. is an integral part of system design. Any process industry has so much of piping, that some of these designs have been standardized to a large extent and one picks from available standard designs rather than designing from scratch every time. Even then the basics of design should not be overlooked.

Designing structures that we see in any chemical process plant is extremely involved. These designs have to bear with harsh process conditions inside and a more and more safety conscious society outside. These are also designs, which do not have the luxury of full dress rehearsals. They must be right in the first attempt. Always keeping the theory in mind and

scrupulously following the codes and standards whenever they are available is the only way for designers dealing with the least forgiving technology, the Chemical Process Technology.

Table 1: SOME DESIGN FORMULAE (to calculate thickness)

Shape	Theory	Code	
		Based on ID	**Based on OD**
Sphere	$t = \frac{P.\,Ri}{2\,Sa}$	$\frac{P.\,Ri}{2\,Sa\,E - 0.2\,P}$	put $Ri = Ro - t$ → $\frac{P.\,Ro}{2\,Sa\,E + 0.8\,P}$
Cylinder	$t = \frac{P.\,Ri}{Sa}$	$\frac{P.\,Ri}{Sa\,E - 0.6\,P}$	$\frac{P.\,Ro}{Sa\,E + 0.4\,P}$

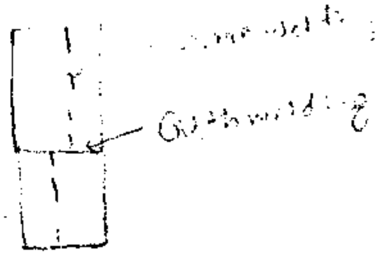
i – inner dimension
o – outer dimension
E – weld joint efficiency

P = Design Pressure (worst operating [illegible])
Sa = allowable stress at design temp.

Ellipsoid — $\frac{P\,Di}{2\,Sa\,E - 0.2\,P}$

Dished $R/r = 16\frac{2}{3}$ $\frac{0.885\,P.\,L.}{Sa\,E - 0.1\,P}$

Conical — $\frac{P\,Di}{2\cos\alpha\,(Sa\,E - 0.6\,P)}$



([illegible] seam welding [illegible]
girth welding = 1
what to take?
Seamwelding

② Incase of calculation of wind load, we give efficiency
① Unless girth welding is half [illegible] of the seam welding, use seamwelding.

[See the page]
(PTO)

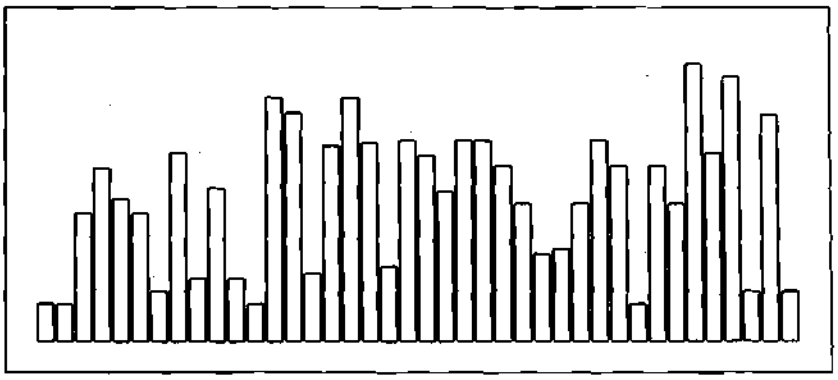
Certificate Course
on

PIPING ENGINEERING

June 12 - 25, 2006

CODES AND STANDARDS

T. N. Gopinath
Consultant



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CODES AND STANDARDS

T. N. GOPINATH

For scientific design of Piping Systems, selection of proper material of construction and to detail out the material specifications, knowledge of Codes and Standards is essential. Standardization can, and does, reduce cost, inconvenience, and confusion that result from unnecessary and undesirable differences in systems, components and procedures. Industry standards are published by professional societies, committees and trade organizations. A code is basically a standard that has been generally accepted by the government. The objective of each code is to ensure public and industrial safety in a particular activity or equipment. Codes are often developed by the same organization that develops standards. These organizations also develop good engineering practices and publish as **Recommended Practices.** The intent of these documents is misunderstood since definition of Codes, Standards and Recommended Practices are not always correctly understood. The following definitions are generally accepted.

CODE

A group of general rules or systematic procedures for design, fabrication, installation and inspection prepared in such a manner that it can be adopted by legal jurisdiction and made into law.

STANDARDS

Documents prepared by a professional group or committee who are believed to be good and proper engineering practice and which contain mandatory requirements. The users are responsible for the correct application of the same. Compliance with a standard does not itself confer immunity from legal obligation.

RECOMMENDED PRACTICES

Documents prepared by professional group or committee indicating good engineering practices but which are optional.

Companies also develop Guides in order to have consistency in the documentation. These cover various engineering methods, which are considered good practices, without specific recommendation or requirements.

Codes and Standards as well as being regulations, might be considered as "design aids" since they provide guidance from experts.

Each country has its own Codes and Standards. On global basis, American National standards are undoubtedly the most widely used and compliance with those requirements are accepted world over. In India, other than American standards, British standards and Indian standards are also used for the design and selection of equipment and piping systems. The major organizations for standards are;

MAJOR ORGANISATION FOR STANDARDS

S. No.	Country	Organization	Abbreviation
1	United States	American National Standards Institute	ANSI
2	Canada	Standard Council of Canada	SCC
3	France	Association Francaise	AFNOR
4	United Kingdom	British Standards Institute	BSI
5	Europe	Committee of European Normalization	CEN
6	Germany	Deutsches Institute Fur Normung	DIN
7	Japan	Japanese Industrial Standards Committee	JISC
8	India	Bureau Of Indian Standards	BIS
9	Worldwide	International Organization for Standards	ISO

ISO is a worldwide federation of national standards bodies from some 100 countries, one from each country. (BS → BSEN British and European Norms)

1.0 AMERICAN STANDARDS

Not all American standards are issued directly by American National Standards Institute. The material standards are covered under ASTM (American Society for Testing and Materials) and dimensional standards under ANSI (American National Standards Institute). Most of these standards are adapted by ASME (American Society of Mechanical Engineers).

The American Standards referred by Piping Engineers are mainly the standards by:

1.1 The American Petroleum Institute (API)

1.2 The American Iron and Steel Institute (AISI)

1.3 The American National Standards Institute (ANSI)

1.4 The American Society for Testing and Materials (ASTM).

1.5 The American Welding Society (AWS).

1.6 The American Water Works Association (AWWA).

1.7 The Manufacturers Standardization Society of Valves and Fitting Industry – Standard Practices (MSS-SP)

1.8 The American Society of Mechanical Engineers (ASME)

1.1 API STANDARDS

The generally referred API standards by the Piping Engineers are :

1)	API 5L	-	Specification for Line Pipe
2)	API 6D/ ISO 14313	-	Pipe line Valves, End closures, Connectors and Swivels.
3)	API 6F	-	Recommended Practice for Fire Test for valves.
4)	ANSI/API RP 574	-	Inspection Practices for Piping Components.
5)	API 593	-	Ductile Iron Plug Valves – flanged ends.
6)	API 598	-	Valve Inspection and Test.
7)	API 600/ ISO 10434	-	Steel Gate Valves
8)	ANSI/API 602/ ISO 15761	-	Compact Design carbon steel Gate, Globe and Check valves ≤ 100 NB.
9)	API 603	-	Corrosion Resistant Gate Valves
10)	API 604	-	Ductile Iron Gate Valves – flanged ends.
11)	API 607	-	Fire test for soft-seated ball valves
12)	API 609	-	Butterfly valves
13)	API 1104	-	Standard for welding pipeline and facilities.
14)	API 609	-	Butterfly Valves – Double flanged, Lug and Wafer Type.
15)	ANSI/ API RP 621	-	Reconditioning of Metallic Gate, globe & Check Valves.
16)	API RP 941	-	Steel for Hydrogen Service at Elevated Temperatures & Pressures.
17)	API RP 1102	-	Steel Pipeline Crossing Rail Roads & Highways.
18)	API 1104	-	Standard for Welding Pipeline and facilities.

19) API RP 1110 - Pressure testing of liquid petroleum lines.

1.2 AISI STANDARDS

The American Iron and Steel Institute Standards specifies the material by its chemical and physical properties. When specific mode of manufacture of the element is not the concern, then the material can be identified by the AISI standards. The most commonly used AISI specifications are:

1) AISI 410 - 13% Chromium Alloy Steel

2) AISI 304 - 18/8 Austenitic Stainless Steel

3) AISI 316 - 18/8/3 Austenitic Stainless Steel

1.3 ANSI STANDARDS

The American National Standards Institute's standards used in the design of the Piping Systems are as listed. In 1978,ANSI B31 committee was reorganized as ASME Code for Pressure Piping B31 committee. Subsequently the code designation was changed. ASME B31 Code for pressure piping is at present a non-mandatory code in USA, though they are adopted as legal requirement.

1) ASME B 31.1 - Power Piping

2) ASME B 31.2 - Fuel Gas Piping

3) ASME B 31.3 - Process Piping

4) ASME B 31.4 - Pipeline Transportation System ~~for liquid~~ hydrocarbon and other Liquids

5) ASME B 31.5 - Refrigeration Piping

6) ASME B 31.8 - Gas Transmission and Distribution Piping Systems.

7) ASME B 31.9 - Building Services Piping

8) ASME B 31.11 - Slurry Transportation Piping Systems

9) ASME B 31.G - Manual for determining the remaining strength of corroded piping - A supplement to ASME B31.

Of the above, the most commonly used code is ASME B 31.3. Refineries and chemical plants are designed based on the same. All power plants are designed as per ASME B31.1.

ASME B31.7 – Nuclear [illegible]
exchanged by ASME – section – 3.

SELECTION OF DESIGN CODE

Unless agreement is specifically made between the contracting parties to use another issue or the regulatory body having jurisdiction imposes the use of another issue, the latest edition and addenda issued 6 months prior to the original contract date will hold good for the first phase of the *completion of work and initial operation*.

It is the responsibility of the user to select the Code Section, which most nearly applies to a proposed piping installation.

Factors to be considered include:
Technical limitations of the Code Section, jurisdictional requirements and the applicability of other Codes and Standards. All applicable requirements of the selected Code *shall* be met. For some installations, more than one Code Section may apply to different parts of the installation. The user is responsible for imposing requirements *supplementary* to those of the Code if necessary to assure safe piping for the proposed installation.

When no section of the Code specifically covers proposed installation, the user has the discretion to select any section determined to be generally applicable. However, it is cautioned that supplementary requirements to the section chosen may be necessary to provide for safe piping system for the intended application.

The Code sets forth engineering requirements deemed necessary for the safe design and construction of pressure piping. While safety is the basic consideration, this factor alone will not necessarily govern the final specification for any piping installation.

The Code prohibits designs and practices known to be unsafe and contains warnings where caution, but not prohibition, is warranted. **The designer is cautioned that the Code *is not a design handbook*; it does not do away with the need for the engineer or competent engineering judgement.**

Other major ANSI / ASME dimensional standards referred for the piping elements are:

1) ASME B 1.1 - Unified Inch Screw Threads

2) ASME B 1.20.1 - Pipe Threads general purpose – (Ex ANSI B2.1)

3) ASME B 16.1 - Cast Iron Pipe Flanges and Flanged Fittings

4) ASME B 16.3 - Malleable Iron Threaded Fittings.

5)	ASME B 16.4	-	Cast Iron Threaded Fittings
6)	ASME B 16.5	-	Steel Pipe flanges and Flanged Fittings
7)	ASME B 16.9	-	Steel Butt welding Fittings
8)	ASME B 16.10	-	Face to face and end to end dimensions of Valves
9)	ASME B 16.11	-	Forged steel Socket welding and Threaded fittings
10)	ASME B 16.20	-	Metallic Gaskets for pipe flanges – ring joint, spiral wound and jacketed flanges
11)	ASME B 16.21	-	Non-Metallic Gasket for pipe flanges
12)	ASME B 16.25	-	Butt Welding Ends
13)	ASME B 16.28	-	Short Radius Elbows and Returns
14)	ASME B 16.34	-	Steel Valves, flanged and butt welding ends.
15)	ASME B 16.42	-	Ductile Iron Pipe Flanges & Flanged Fittings –Class 150 and 300
16)	ASME B 16.47	-	Large Diameter Steel Flanges – NPS 26 to 60
17)	ASME B 16.49	-	Factory made Induction bends
18)	ASME B 18.2 1 & 2	-	Square and hexagonal head Bolts and Nuts – (in & mm)
19)	ASME B 36.10	-	Welded and seamless Wrought Steel Pipes
20)	ASME B 36.19	-	Welded and Seamless Austenitic Stainless Steel Pipes.

1.4 ASTM STANDARDS

ASTM standards consist of 16 sections on definitions and classifications of materials of construction and test methods. Most of the ASTM standards are adapted by ASME and are specified in ASME Section II. The Section II has four parts.

1.4.1	Part-A	-	Ferrous materials specifications
1.4.2	Part-B	-	Non-ferrous metals specification
1.4.3	Part-C	-	Specification for welding materials

ASTM A 106

(ASME) SA 106

1.4.4 Part-D - Properties of materials.

In Part-II, the materials are listed in the Index based on the available forms such as plates, castings, tubes, etc. and also on the numerical index.

The selection of ASTM specification depends upon the type of manufacture, form of material, its mechanical strength and the corrosion properties.

The specification number is given on Alphabetical prefix, 'A' for Ferrous materials and 'B' for Non-ferrous materials.

ASTM also specifies standard practice for numbering metal and alloys as Unified Numbering System.

UNIFIED NUMBERING SYSTEM (UNS)

The UNS number itself is not a specification, since it establishes no requirements for form, condition, quality etc. It is a unified identification of metals and alloys for which controlling limits have been established in specification elsewhere.

The UNS provides means of correlating many naturally used numbering systems currently administered by Societies, trade associations, individual users and producers of metals and alloys, thereby avoiding confusion caused by use of more than one identification number for the same material and by the opposite situation of having the same number assigned to two different materials.

UNS establishes 18 series numbers of metals and alloys. Each UNS number consists of a single letter prefix followed by five digits. In most cases the alphabet is suggestive of the family of the metal identified.

1. A00001 - A 99999 - Aluminium and Aluminium alloys
2. C00001 - C 99999 - Copper and Copper alloys
3. E00001 - E 99999 - Rare earth and rare earth like metals and alloys
4. L00001 - L 99999 - Low melting metals and alloys
5. M00001 - M 99999 - Miscellaneous non-ferrous metals and alloys
6. N00001 - N 99999 - Nickel and Nickel alloys
7. P00001 - P 99999 - Precious metals and alloys
8. R00001 - R 99999 - Reactive and Refractory metals and alloys
9. Z00001 - Z 99999 - Zinc and Zinc alloys
10. D00001 - D 99999 - Specified mechanical properties of Steels

11. F00001 - F 99999 - Cast Iron and Cast Steels
12. G00001 - G 99999 - AISI and SAE Carbon and Alloy steels
13. H00001 - H 99999 - AISI H Steels
14. J00001 - J 99999 - Cast Steels
15. K00001 - K 99999 - Miscellaneous Steels and Ferrous alloys
16. S00001 - S 99999 - Stainless Steels
17. T00001 - T 99999 - Tool Steels
18. W00001 - W99999 - Welding Filler Metals and Electrodes

1.5 AWS STANDARDS

The American Welding Society (AWS) standards provide information on welding fundamentals; weld design, welders' training qualification, testing and inspection of welds and guidance on the application and use of welds. Individual electrode manufacturers have given their own brand names for the various electrodes and are sold under these names.

1.6 AWWA STANDARDS

The American Water Works Association (AWWA) standards refer to the piping elements required for low-pressure water services. These are less stringent than other standards. Valves, flanges, etc. required for large diameter water pipelines are covered under this standard and are referred rarely by CPI Piping Engineers.

1) C-500 - Gate Valves for water & sewage system
2) C-510 - Cast Iron Sluice Gates
3) C-504 - Rubber Seated Butterfly Valves
4) C-507 - Ball valves 6" – 48"
5) C-508 - Swing Check Valves 2" – 24"
6) C-509 - Resilient Seated Gate Valves for water & sewage

1.7 MSS-SP STANDARDS

In addition to the above standards and material codes, there are standard practices followed by manufacturers. These are published as advisory standards and are widely

followed. A large number of MSS Practices have been approved by the ANSI & ANSI Standards published by others. In order to maintain a single source of authoritative information, the MSS withdraws those Standard Practices in such cases. The most common MSS-SP standards referred for piping are:

1) MSS-SP-6 - Standard Finishes for Contact Surface for Flanges

2) MSS-SP-25 - Standard Marking System for Valves, Fittings Flanges

3) MSS-SP-42 - Class 150 Corrosion Resistant Gate, Globe and Check Valves.

4) MSS-SP-43 - Wrought Stainless Steel Buttweld Fittings

5) MSS-SP-56 - Pipe Hanger Supports: Materials, Design and Manufacture

6) MSS-SP-61 - Pressure testing of Steel Valves

7) MSS-SP-67 - Butterfly Valves

8) MSS-SP-68 - High Pressure Offseat Design Butterfly Valves

9) MSS-SP-69 - Pipe Hangers and Supports: Selection and application

10) MSS-SP-70 - Cast Iron Gate Valves

11) MSS-SP-71 - Cast Iron Check Valves

12) MSS-SP-72 - Ball Valves

13) MSS-SP-75 - High test wrought buttwelding fittings

14) MSS-SP-78 - Cast Iron Plug Valves

15) MSS-SP-80 - Bronze Gate, Globe and Check Valves

16) MSS-SP-81 - Stainless Steel Bonnetless Knife Gate Valves

17) MSS-SP-83 - Pipe Unions

18) MSS-SP-85 - Cast Iron Globe Valves

19) MSS-SP-88 - Diaphragm Type Valves

20)	MSS-SP-89	-	Pipe Hangers and Supports: Fabrication and installation practices.
21)	MSS-SP-90	-	Pipe Hangers and Supports: Guidelines on terminology
22)	MSS-SP-92	-	MSS Valve user guide
23)	MSS-SP-108	-	Resilient Seated Eccentric CI Plug Valves.
24)	MSS-SP-115	-	Excess Flow Valves for Natural Gas Service.
25)	MSS-SP-122	-	Plastic Industrial Ball Valves

2.0 BRITISH STANDARDS

In many instances, it is possible to find a British Standard, which may be substituted for American Standards. Now the Community for European Normalization is issuing standards replacing different standards in force in the European countries. Accordingly lot of BS and DIN standards are getting replaced by CEN standards.

There are certain British Standards referred to by Indian Manufacturers for the construction of piping elements such as valves. The most commonly referred British standards in the Piping Industry are:

1)	BS 10	-	Flanges
2)	BS 916	-	Black Bolts, Nuts and Screws (obsolescent)
3)	BS 970	-	Steel for forging, bars, rods, valve steel, etc.
4)	BS 1306	-	Copper and Copper alloy pressure piping system
5)	BS 1414	-	Gate Valves for Petroleum Industry (withdrawn & replaced by BSEN ISO – 10434)
6)	BS 1560	-	Steel Pipe Flanges (class designated)
7)	BS 1600	-	Dimensions of Steel Pipes
8)	BS 1640	-	Butt Welding Fittings
9)	BS 1868	-	Steel Check Valves for Petroleum Industry

10) BS 1873	-	Steel Globe & Check Valves for Petroleum Industry
11) BS 1965	-	Butt welding pipe fittings
12) BS 2080	-	Face to Face / End to End dimensions of Valves (obsolescent)
13) BS 2598	-	Glass Pipelines and Fittings
14) BS 3059	-	Boiler and Super Heater tubes
15) BS 3063	-	Dimensions of Gaskets for pipe flanges (obsolescent)
16) BS 3293	-	C.S. Flanges 26"-48" NB
17) BS 3381	-	Metallic Spiral Wound Gaskets
18) BS 3600	-	Dimensions of Welded and Seamless Pipes & Tubes.
19) BS 3601	-	C.S. Pipes & Tubes for pressure purposes at room temperature
20) BS 3602	-	C.S. Pipes & Tubes for pressure purposes at high temperature
21) BS 3603	-	C.S. and Alloy steel Pipes & Tubes for pressure purposes at low temperature.
22) BS 3604	-	Alloy steel Pipes & Tubes for high temperature
23) BS 3605	-	S.S. Pipes & Tubes for pressure purposes
24) BS 3799	-	SW/Screwed Fittings
25) BS 3974	-	Pipe hangers, Slides & Roller type Supports.
26) BS 4346	-	PVC pressure Pipe – joints & Fittings
27) BS 4504	-	Steel, CI & Copper alloy Flanges (PN designated).
28) BS 5150	-	CI Wedge and Double Disc Gate Valves for general purposes

29)	BS 5151	-	CI Gate (parallel slide) valves for general purposes
30)	BS 5152	-	CI Globe & Check valves for general purposes.
31)	BS 5153	-	CI Check valves for general purposes.
32)	BS 5154	-	Copper alloy Gate, Globe, Check valves
33)	BS 5155	-	Butterfly valves (withdrawn & replaced by BSEN 593)
34)	BS 5156	-	Diaphragm valves for general purposes (replaced by BSEN 13397)
35)	BS 5158	-	CI and CS Plug valves for general purposes
36)	BS 5159	-	CI and CS Ball valves for general purposes
37)	BS 5160	-	Flanged steel Globe and Check valves for general purposes
38)	BS 5163	-	Double flanged Cast Iron wedge gate valves for water works purposes.
39)	BS 5351	-	Steel Ball Valves for petroleum industries (replaced by ISO 17292)
40)	BS 5352	-	Steel Gate, Globe, Check Valves < 2" NB (Withdrawn & replaced by BSEN ISO – 15761)
41)	BS 5353	-	Specification for Plug Valves
42)	BS 5391	-	Specification for ABS Pressure Pipes
43)	BS 5392	-	Specification for ABS Fittings
44)	BS 5433	-	Specification for underground Stop Valves for water services
45)	BS 5480	-	Specification for GRP Pipes and Fittings
46)	BS 6364	-	Specification for Valves for cryogenic services
47)	BS 6755	-	Testing of valves (replaced by BSEN 12266 & ISO 10497

48)	BS 7159	-	Code of Practice, Design & Construction GRP Piping Systems
49)	BS 7291	-	Specification for thermoplastic pipe and pipe fittings (PB, PE, CPVC)
50)	BS 8010	-	Code of Practice for Pipelines

3.0 INDIAN STANDARDS

Bureau of Indian Standards (**BIS**) have so far not developed an Indian standard for the design of Piping Systems. Hence, ANSI standards ASME B 31.1/31.3 are widely referred for the design. These standards also accept materials covered in other standards. Unlike American Standards, Indian Standards cover dimensions and material specifications under the same standard. There are also no groupings done based on the series/branch of engineering as well. Some of the most commonly referred Indian Standards by the Piping Engineers are:

1)	IS – 210	-	Grey Iron Castings
2)	IS – 226	-	Structural Steel (superseded by IS 2062)
3)	IS - 554	-	Dimensions of Pipe Threads
4)	IS – 778	-	Specification for Copper Alloy Gate, Globe and Check Valves.
5)	IS 1239	-	Specification for Mild Steel Tubes and Fittings. Part I & Part II (Thickness as Light, medium, heavy)
6)	IS 1363	-	Hexagonal bolts, screws and nuts – Grade C
7)	IS 1364	-	Hexagonal bolts, screws and nuts – Grade A & B
8)	IS 1367	-	Technical supply conditions for threaded steel fasteners
9)	IS 1536	-	Centrifugally Cast Iron Pipes
10)	IS 1537	-	Vertically Cast Iron Pipes
11)	IS 1538	-	Cast Iron Fittings

12)	IS 1870	-	Comparison of Indian and Overseas Standards
13)	IS 1879	-	Malleable Iron Pipe Fittings
14)	IS 1978	-	Line Pipe
15)	IS 1979	-	High Test Line Pipe
16)	IS 2002	-	Steel Plates – Boiler Quality
17)	IS 2016	-	Plain Washers
18)	IS 2041	-	Steel Plates for pressure vessel used at moderate and low temperature
19)	IS 2062	-	Steel for general structural purposes
20)	IS 2379	-	Colour code for identification of pipelines
21)	IS 2712	-	Compressed Asbestos Fibre jointing
22)	IS 2825	-	Code for unfired pressure vessels
23)	IS 3076	-	Specification for LDPE Pipes
24)	IS 3114	-	Code of Practice for laying CI Pipes
25)	IS 3516	-	CI Flanges and Flanged Fittings for Petroleum Industry
26)	IS 3589	-	Seamless or ERW Pipes (150 NB to 2540 NB) (Thickness in actual dimension)
27)	IS 4038	-	Specification for Foot Valves
28)	IS 4179	-	Sizes for Pressure Vessels and leading dimensions
29)	IS 4853	-	Radiographic examination of butt weld joints in pipes.
30)	IS 4864 to IS 4870	-	Shell Flanges for vessels and equipment
31)	IS 4984	-	Specification for HDPE Pipes for water supply
32)	IS 4985	-	Specification for PVC Pipes
33)	IS 5312	-	Specification for Swing Check Valves

34)	IS 5572	-	Classification of hazardous area for electrical installation
35)	IS 5822	-	Code of practice for laying welded steel pipes
36)	IS 6157	-	Valve Inspection and Test
37)	IS 6286	-	Seamless and Welded Pipe for subzero temperature
38)	IS 6392	-	Steel Pipe Flanges
39)	IS 6630	-	Seamless Alloy Steel Pipes for high temperature services
40)	IS 6913	-	Stainless steel tubes for food and beverage industry
41)	IS 7181	-	Horizontally Cast iron pipes
42)	IS 7719	-	Metallic spiral wound gaskets
43)	IS 7806	-	SS Castings
44)	IS 7899	-	Alloy steel castings for pressure services
45)	IS 8008	-	Specification for moulded HDPE Fittings
46)	IS 8360	-	Specification for fabricated HDPE Fittings
47)	IS 9890	-	Ball Valves for general purposes
48)	IS 10124	-	PVC fittings for potable water supplies
49)	IS 10221	-	Code of practice for coating and wrapping of underground MS pipelines
50)	IS 10592	-	Eye wash and safety showers
51)	IS 10605	-	Steel Globe Valves for Petroleum Industries
52)	IS 10611	-	Steel Gate Valves for Petroleum Industries
53)	IS 10711	-	Size of drawing sheets
54)	IS 11323	-	Steel Gate Valves for Marine Piping systems

55)	IS 11665	-	Technical drawings Title Block
56)	IS 11790	-	Code of practice for preparation of Butt welding ends for valves, flanges and fittings.
57)	IS 11791	-	Diaphragm Valves for general purposes
58)	IS 11792	-	Steel Ball Valves for Petroleum Industries
59)	IS 14164	-	Code of practice for insulation
60)	IS 14333	-	HDPE pipes for sewerage purposes
61)	IS 14845	-	Resilent seated Cast Iron Air Relief Valves
62)	IS 14846	-	Sluice valves for water works 50-1200 mm

There are certain other international standards also referred in the piping industry. They are the DIN standards of Germany and the JISC standards of Japan. DIN standards are more popular and equivalent British and Indian standards are also available for certain piping elements.

Periodic review of the standards by the committee is held and these are revised to incorporate the modified features based on the results of research and feedback from the industry. Although some technological lags are unavoidable, these are kept minimum by those updations. Hence, it is necessary that the latest editions of the codes and standards are referred for the design and year of publication also to be indicated along with.

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Certificate Course
on
PIPING ENGINEERING

June 12 - 25, 2006

PIPING ELEMENTS

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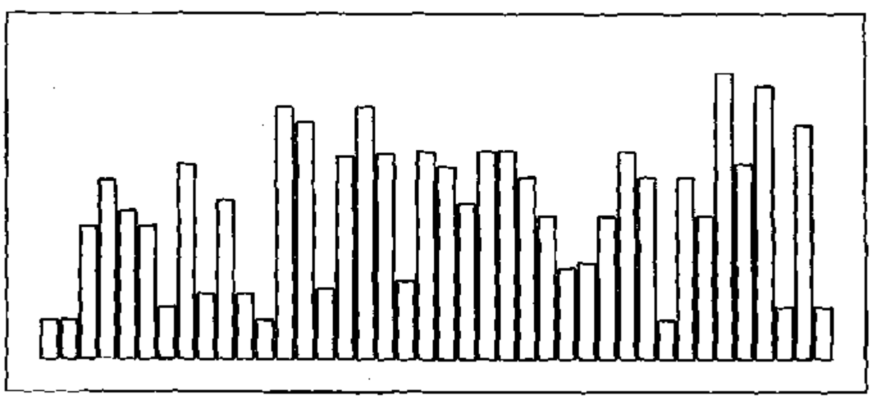
Consultant



Organized by

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PIPING ELEMENTS

T. N. GOPINATH

One of the major tasks in any process industry is the transportation of materials often in fluid form from one place to another. The most commonly adopted method for the same is to force the fluid through the piping system. The piping system is the inter-connected piping subject to the same set of design conditions. The piping system involves not only pipes but also the fittings, valves and other specialties. These items are known as piping components. Code specifies the piping components as mechanical elements suitable for joining or assembly into pressure-tight fluid-containing piping systems. Components include

1.0 Pipes
2.0 Fittings
3.0 Flanges
4.0 Gaskets
5.0 Bolting
6.0 Valves
7.0 Specialties

Piping element is defined as any material or work required to plan and install the piping system. Elements of piping include design specifications, materials, components, supports, fabrication, inspection and testing.

Piping elements should, so far as practicable, conform to the specification and standards listed in the code referred for design. Unapproved elements may also be used provided they are qualified for use as set forth in applicable chapters of the code.

Piping specification is a document specifying each of the components. Different material specifications are segregated in different "Piping Class". Identification of the "Piping Classes" depends on each Designer, and the logic he/she adopts.

MATERIAL SELECTION OF PIPING COMPONENTS

The first thing to be considered is the selection of suitable material for the service. The selection of piping material requires knowledge of corrosion properties, strength and engineering characteristics, relative cost and availability.

The main process considerations in the material selection are the corrosion properties of the fluid, the pressure temperature conditions of the service and the nature of the service.

The Piping Designer selects/designs the piping components based on the mechanical properties such as the following.

a. Yield strength
b. Ultimate strength
c. Percentage elongation
d. Impact strength
e. Creep-rupture strength
f. Fatigue endurance strength

Based on the material of construction piping elements could be classified as shown in Fig.1. 1

The basic material or the generic material of construction is specified by the Process Licenser for the process fluids. The Piping Engineer is expected to detail out the same based on the Codes and Standards. The material of construction for the utilities will be selected by the Piping Engineer based on the service conditions.

The Piping Design Criteria originates from the Line List, which specifies design conditions with respect to pressure and temperature.

In absence of this data, the Piping Engineer considers the following for strength calculations.

PTFE temp. limit ~200° max

MS: Mild Steel

not used in Refineries

Ni, Cr, Mo → added to CI for strength

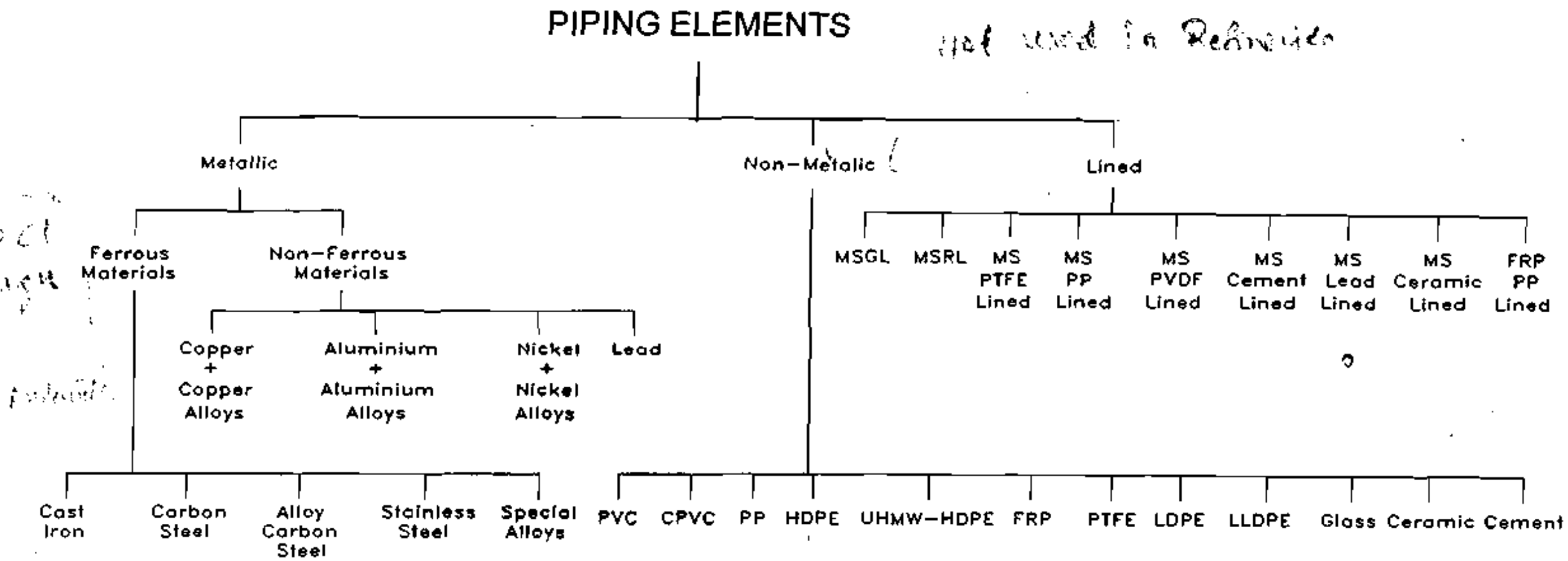


FIGURE 1.1

PIPING ELEMENTS

CLASSIFICATION BASED ON MATERIAL OF CONSTRUCTION

a) Design Pressure as 10% higher than the maximum anticipated operating pressure.
b) Design Temperature as 25° above the maximum anticipated operating temperature.
c) When operating temperature is 15°C and below, the design temperature as the anticipated minimum operating temperature.

The design should meet the requirements of the relevant code.

The material used shall be in accordance with latest revision of standards. If ASTM materials are used, then the materials adapted by ASME should be preferred.

The selection of materials in general shall follow the norms below: (The basis in the design code governs.)

a) Carbon steel shall be used up to 800°F (425°C).
b) Low temperature steel shall be used below - 20°F (- 29° C)
c) Alloy carbon steel shall be used above 800°F (425°C).
d) For corrosive fluids, recommendations from the Process Licensor to be followed.

1.0 PIPES

1.1 General

Pipe can be defined as a pressure tight cylinder used to convey a fluid.

The word "pipe" is used as distinguished from "tube" to apply to tubular products of dimensions commonly used for piping systems. The pipe dimensions of sizes 12 inch (300 mm) and smaller have outside diameter numerically larger than corresponding sizes. In contrast, the outside diameter of tubes is numerically identical to the size number for all sizes.

The Pipes and Tubes can be compared on the following lines:

Tube

1. Lower thickness and higher ductility permits rolling into coils without high differential stress between inside and outside of coil.

2. Specified by outside diameter and actual thickness in mm/mch or wire gauges.

3. Uniform thickness means less chance of tube failure due to hot spots.

4. Low roughness factor and lower pressure drop.

5. Normally used in heat exchangers & coils for heat transfer.

6. Limitation in sizes.

Pipe

1. Lower ductility makes it unsuitable to coil. Due to higher Moment of Inertia larger bending moment is required for the same radius. This means larger residual stress.

2. Specified by Nominal Bore and thickness by Schedule.

3. Variation in thickness can cause hot spots and consequent failures.

4. Higher roughness factor and high pressure drop.

5. Normally used in straight length for fluid transfer.

6. No limitation.

1.2 Size

The size of the pipe is identified by the NOMINAL BORE or the NOMINAL

PIPE SIZE. The manufacture of pipe is based on outside diameter, which is standardized. The O D was originally selected so that pipe with standard wall thickness, which was typical of that period, would have an internal diameter approximately equal to the nominal size.
In American standard, the pipes are covered under

a) ASME B 36.10 - Welded and Seamless Wrought Steel Pipe
b) ASME B 36.19 - Stainless Steel Pipe

The nominal bore and the corresponding outside diameters specified therein are as given in the accompanying table. American standards have not metricated the pipe sizes and the equivalent metric sizes widely followed are also noted along with. However, the latest revisions of these standards include the SI metric dimensions for OD, thickness and unit weight.

As regards the non-metallic and lined piping systems, the thickness of pipe and/or lining are not covered under any of the above standards. These are as per the relevant ASTM standards. For certain plastic pipes, Indian Standards are also available.

Pipes are designated by its **Nominal Bore (NB)** Eg : 2" NB or 50 mm NB.Further pipe can also be designated as

Nominal Pipe Size (NPS) Which is a dimensionless designator in USCS. If indicates standard pipe size when followed by specific size designation number without inch symbol. Eg : NPS 2, NPS12 etc.

Nominal Diameter (DN), again a dimensionless designator in metric system If indicates standard size designation number without millimeter symbol. Eg : DN 50, DN 300 etc.

 designations, not actual conversions!

Pipe Size NB (Inch) designation	Eq. Metric Pipe Size NB (mm)	Outside Dia (Inch)	Outside Dia (mm)
1/8	6	0.405	10.3
1/4	8	0.540	13.7
3/8	10	0.675	17.1
1/2	15	0.840	21.3
3/4	20	1.050	26.7
1	25	1.315	33.4
*1¼	32	1.660	42.2
1½	40	1.900	48.3
2	50	2.375	60.3
*2½	65	2.875	73.0
3	80	3.500	88.9
*3½	90	4.000	101.6
4	100	4.500	114.3
*5	125	5.563	141.3
6	150	6.625	168.3
8	200	8.625	219.1
10	250	10.750	273.0
12	300	12.750	323.9
14	350	14.000	355.6
16	400	16.000	406.4
18	450	18.000	457.2
20	500	20.000	508.0
*22	550	22.000	558.8
24	600	24.000	609.6

(½" NB is min size in almost all industry)

(1" NB doesn't indicate 25.4 mm, it's just designation)

1.3 Wall Thickness

Prior to ASME B 36.10 & ASME B 36.19 became effective, the pipes were manufactured as per the Iron Pipe Standard (IPS) with wall thickness designations Standard (STD), Extra Strong (XS) and Double Extra Strong (XXS).

Subsequently schedule numbers were added as convenient designations. The pipe thickness is designated by Schedule Number and the corresponding thickness is specified in the standard ASME B 36.10 for carbon steel pipes & ASME B 36.19 for stainless steel pipes.

Stainless steel pipes are available in schedule 5S, 10S, 40S and 80S whereas carbon steel pipes are available in schedule 10, 20, 30, 40, 60, 80, 100, 120, 140, 160, STD, XS, XXS.

Thickness Standard and Schedule 40 are identical for nominal pipe sizes upto 10 inch (250 mm) inclusive. All larger sizes of STD have 3/8-inch (10 mm) wall thickness. Extra strong and Schedule 80 are identical of nominal pipe sizes upto 8 inch (200 mm) inclusive. All larger sizes of Extra strong have ½ inch (12.7 mm) wall thickness. The thickness Double Extra Strong is more than Schedule 160 in pipe sizes upto 6 inch (150 mm) NB. This thickness is specified for pipe up to 12 inch (300 mm) NB. For 12 inch (300 mm) NB the thickness matches to that of Schedule120 and for 10 inch (250 mm)

generally (except Plate fabricated SS)
±12.5% tolerance = acceptable

NB it is Schedule 140.
The figures indicated in these standards are the nominal thickness and mill tolerance of ± 12.5% is applicable to those values.

Generally the thickness specified by schedule numbers of B36.10 and B36.19 match except in the followings:

10" ~~SCH80~~/SCH80S
12" SCH40/SCH40S
12" SCH80/SCH80S
14" SCH10/SCH10S
16" SCH10/SCH10S
18" SCH10/SCH10S
20" SCH10/SCH10S
22" SCH10/SCH10S

In Indian Standard IS 1239, the thicknesses of pipes are specified as Light, Medium and Heavy. The medium and heavy pipes are only used for fluid handling. In IS 3589, the thicknesses are specified in actual dimensions in mm.

As regards the non-metallic and lined piping systems, the thickness of pipe and/or lining are not covered under any of the above standards. These are as per the relevant ASTM standards. For certain plastic pipes, Indian Standards are also available.

The pipes are available in standard lengths of 20 feet (6 m).

1.4 Pipe Ends

Based on the material of construction and the pipe to pipe joint, the ends of the pipes are specified as follows.

1.4.1	Beveled ends
1.4.2	Plain ends
1.4.3	Screwed ends
1.4.4	Flanged ends
1.4.5	Spigot/Socket ends
1.4.6	Buttress ends

Beveled ends are specified when pipe to pipe and/or pipe to fittings joints are done by butt welding.

Plain ends are specified when pipe to pipe and/or pipe to fittings joints are done by fillet welding.

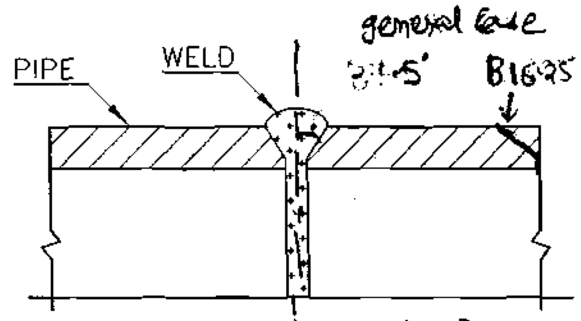
Screwed joints are specified when pipe to pipe and/or pipe to fittings joints are done by threaded connections.

Flanged ends are specified to provide bolted connections between pipes and between pipes and/or fittings.

Spigot/ Socket ends are specified when lead caulked/cemented joints are provided between pipes and between pipes and fittings.

Buttress ends are used in glass piping and are joined by bolting with the use of backing flanges. (Bevel ends)

1.4.1 BUTT WELD PIPE JOINTS



(can be checked by NDT)

Advantages

a) Most practical way of joining big bore piping
b) Reliable leak proof joint
c) Joint can be radiographed

Disadvantages

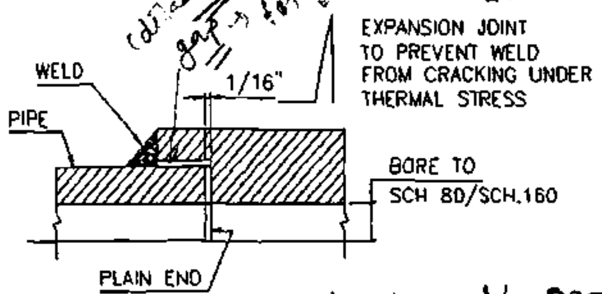
a) Weld intrusion will affect flow
b) End preparation is necessary

general term
1/2 - 1 1/2 - small bore
2" - above - large bore

1.4.2 SOCKET WELD PIPE JOINTS

ends

gaps to avoid cracks



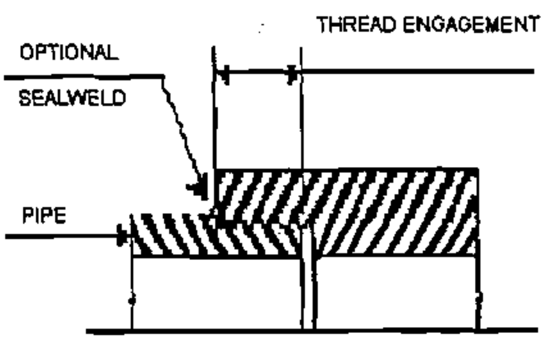
Advantages (may be checked by DPT & MPT)

a) Easier Alignment than butt welding
b) No weld metal intrusion into bore

Disadvantages

a) The 1/16"(1.5 mm) recess pockets liquid
b) Use not permitted by code if Severe Erosion or Crevice Corrosion is anticipated.

1.4.3 SCREWED PIPE JOINTS



Advantages

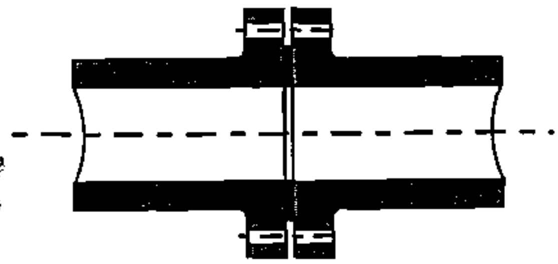
a) Easily made at site
b) Can be used where welding is not permitted due to fire hazard

Disadvantages

a) Joint may leak when not properly sealed
b) Use not permitted by code if severe erosion, crevice corrosion, shock or vibration are anticipated.
c) Strength of pipe is reduced as threads reduce wall thickness
d) Seal welding may be required

e) Code specifies that seal welding shall not be considered to contribute for strength of joint

1.4.4 FLANGED PIPE JOINTS



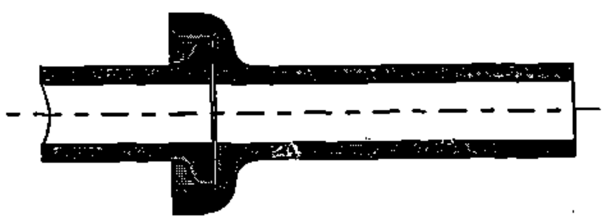
Advantages

a) Can be easily made at site
b) Can be used where welding is not permitted due to material properties or fire hazard.
c) Dismantling is very easy

Disadvantages

a) It is a point of potential leakage
b) Cannot be used when piping is subjected to high bending moment.

1.4.5 SPIGOT SOCKET PIPE JOINTS



Advantages

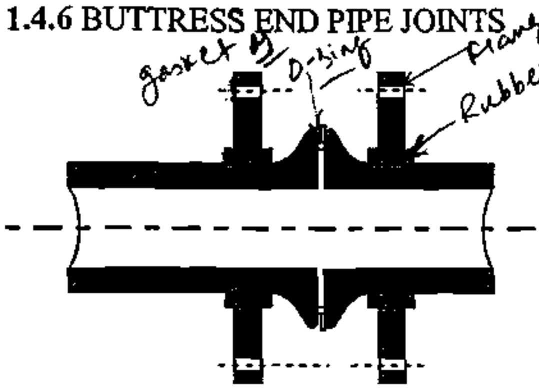
a) Can be easily made at site.
b) Can accept misalignment upto 10^0 at pipe joints.

Disadvantages

a) Suitable for low pressure application.
b) Special configuration at pipe ends required.

near flanged joint supports should be provided to avoid bending moment

1.4.6 BUTTRESS END PIPE JOINTS



Used only for glass piping and not capable to hold high pressure.

1.5 Types Of Pipes

Based on the method of manufacture pipes could be classified as

1.5.1 Welded

a) Electric Resistance Welded (ERW)

Pipes having longitudinal butt joint wherein coalescence is produced by the heat obtained from resistance of the pipe to flow of electric current in a circuit of which the pipe is a part, and by application of pressure.

b) Furnace Butt Welded, Continuous Welded

Pipes having longitudinal weld joints forge welded by mechanical pressure developed in passing the hot-formed and edge-heated skelp through round pass weld rolls.

c) Electric Fusion Welded (EFW)

Pipes having longitudinal butt joint wherein coalescence is produced in the preformed tube by manual or automatic electric arc welding. Weld may be single or double.

d) Double Submerged-Arc Welded

Pipes having longitudinal butt joint produced by at least two passes, one of which is on the inside of the pipe. Coalescence is produced by heating with an electric arc or arcs between the bare metal electrode or electrodes and the pipe. Pressure is not used and filler material is obtained from electrode.

e) Spiral Welded

Pipes having helical seam with either a butt, lap, lock-seam joint which is welded using either an electric resistance, electric fusion or double submerged arc welding process.

1.5.2 Seamless

Pipes produced by piercing a billet followed by rolling or drawing or both.

The most commonly used material standards for the pipes are listed below:

1.6 Pipe Materials

1.	ASTM A 53	Welded and Seamless Steel Pipe, Black and Galvanized
2.	ASTM A106	Seamless CS Pipe for High Temp. Services
3.	ASTM A120	Black and Hot Dipped Zinc coated (Galvanized) welded and seamless pipe for ordinary use
4.	ASTM A134	Electric fusion welded steel plate pipe (Sizes ≥ 16" NB)
5.	ASTM A135	Electric resistance welded pipe
6.	ASTM A155	Electric fusion welded steel pipe for high temperature service
7.	ASTM A312	Seamless and welded austenitic stainless steel pipes
8.	ASTM A333	Seamless and welded steel pipe for low temperature service
9.	ASTM A335	Seamless ferritic alloy steel pipe for high temperature service
10.	ASTM A358	Electric fusion welded austenitic chrome-nickel steel pipe for

UTS { Gr-A 48,000 PSI, Gr-B - 60,000 PSI

ASME B31·3

$AS = \frac{UTS}{3} \approx \frac{2}{3} YS$ | (Max. of both

		high temperature service
11. ASTM	A369	Carbon and ferritic alloy steel forged and bored for high temperature service
12. ASTM	A376	Seamless austenitic steel pipe for high temperature central station service
13. ASTM	A409	Welded large diameter austenitic steel pipe for corrosive or high temperature service
14. ASTM	A426	Centrifugally cast ferritic alloy steel pipe for high temperature service
15. ASTM	A430	Austenitic steel forged and bored pipe for high temperature service
16. ASTM	A451	Centrifugally cast austenitic steel pipe for high temperature service
17. ASTM	A452	Centrifugally cast austenitic steel cold wrought pipe for high temperature service
18. ASTM	A524	Seamless carbon steel pipe for atmospheric and low temperature services
19. ASTM	A587	Electric welded low carbon steel pipe for the chemical industry
20. ASTM	A660	Centrifugally cast carbon steel pipe for high temperature service
21. ASTM	A671	Electric fusion welded steel pipe for atmospheric and low temperature service (Sizes ≥ 16" NB)
22. ASTM	A672	Electric fusion welded steel pipe for high pressure service at moderate temperature services (Sizes ≥ 16" NB)
23. ASTM	A691	Carbon and alloy steel pipe, electric fusion welded for high pressure service at high temperatures (Sizes ≥ 16" NB)
24. ASTM	A731	Seamless and welded ferritic stainless steel pipe
25. ASTM	A790	Seamless and welded ferritic/ austenitic stainless steel pipe
26. ASTM	A813	Single or double welded austenitic stainless steel pipe
27. ASTM	A814	Cold worked welded austenitic stainless steel pipe
28. ASTM	F1545	Plastic Lined Ferrous Pipe
29. API 5L		Line pipe
30. IS 1239		Steel pipes for general purposes (Sizes ≤ 6" NB)
31. IS 1536		Centrifugally cast iron pipe
32. IS 1537		Vertically cast iron pipe
33. IS 1978		Line pipe
34. IS 1979		High test line pipe
35. IS 3589		Steel pipe for general services
36. IS 4984		HDPE pipe for water service
37. IS 4985		PVC pipe

1.7 Pressure Design

Codes specify the formula to arrive at the required thickness for the pipes to withstand internal/external pressure to which the system is subjected to. Unlike

Low temp → < -20'F

pressure vessels, the pipes and fittings are manufactured to certain standard dimensions.

Hence, it is necessary for the Piping Engineer to select the best suited thickness of the element.

Corrosion allowance, depending on the service to which the system is subjected to and the material of construction, is to be added to the calculated minimum thickness.

The thickness arrived thus is to be compared with the available standard thickness after allowing for the mill tolerance of ±12.5% on the nominal thickness.

1.7.1. THICKNESS OF STRAIGHT PIPE UNDER INTERNAL PRESSURE

ASME B 31.3, the Process Piping Code, in clause 304.1.1 gives minimum thickness as follows:

$$Tm = T + C$$

$$\text{where } T = \frac{PD}{2(SE + PY)}$$

according to latest B31.3

$$\frac{PD}{2(SEW+PY)}$$

where W = weld joint strength Red^n factor

- P = Internal Design gauge pressure
- D = Outside Diameter of pipe
- S = Allowable Stress from Appendix A 1
- E = Joint Quality factor from Table A – 1B
- Y = Coefficient from 304.1.1
- C = C1 + C2
- C1 = Corrosion Allowance
  - = 1.6 mm in general for carbon steel
  - = 0 for stainless steel
- C2 = Depth of thread (used only upto 1½" NB)

The calculated thickness to be corrected to consider the mill tolerance of - 12.5% as

$$Tm = \frac{8}{7}\left(\frac{PD}{2(SE + PY)} + C1\right) + C2$$

The use of the above equation is best illustrated by means of the following example.

Example:

A 12" (300 mm) NB pipe has an internal maximum operating pressure of 500 psig (35 kg/cm^{2}g) and temperature of 675^{0}F. The material of construction of the pipe is seamless carbon steel to ASTM A106 Gr B. The recommended corrosion allowance is 1/8" (3mm). Calculate the thickness of pipe as per ASME B 31.3 and select the proper schedule.

$$Tm = \frac{PD}{2(SE + PY)} + C$$

P = 10% higher than the MWP
= 1.1 x 500 = 550 psig

D = 12.75" (OD of 12" NB pipe)

Design temperature = 675 + 25
= 700^0 F

S = 16500 psi (at 700°F, for A106 Gr.B)
(Refer ASME B 31.3 Appendix 'A' Table A-1)

E = 1 (Joint Quality factor. Refer ASME B31.3, Appendix 'A' Table A-1B)

Y = 0.4 (Refer Table 304.1.1)

C = 0.125" (Specified)

$$Tm = \frac{550 \times 12.75}{2(16500 \times 1 + 550 \times 0.4)} + 0.125$$

= 0.2097" + 0.125"
= 0.335"

Hence, considering the mill tolerance of 12.5%, the nominal thickness for a minimum thickness of 0.335" will be

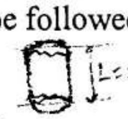
$$t = \frac{0.335}{0.875} = 0.383"$$

In practice we will specify SCH 40 pipe, which has a nominal wall thickness of 0.406" and minimum 0.355" (0.406x0.875).

1.7.2 THICKNESS OF STRAIGHT PIPE UNDER EXTERNAL PRESSURE

The pipe with a large ratio of diameter to wall thickness will collapse under an external pressure which is only a small fraction of internal pressure which it is capable of withstanding.

To determine the wall thickness under external pressure, the procedure outlined in the BPV Code ASME Section VIII Div. 1 UG-28 through UG-30 shall be followed.



L = straight height

Example:

A 6" (150 mm) NB pipe has an external Design Pressure of 400 psig at 750^0 F. The material of construction of pipe is seamless austenitic stainless steel to ASTM A 312 TP 304L. The corrosion allowance is nil. Calculate thickness and select proper schedule.

Refer ASME Section VIII Div.1. UG 28

Assume value of 't' and determine ratios

$$\frac{L}{Do} \quad \text{and} \quad \frac{Do}{t}$$

Do for 6" NB pipe = 6.625"

Assume SCH 5 S pipe

Nominal thickness = 0.109"

Minimum thickness considering negative mill tolerance of 12.5%

t = 0.875 x 0.109 = 0.095" $\left(1 - \frac{12.5}{100}\right)$

Consider, $\frac{L}{Do} = 50$

since L is unspecified.

$$\frac{Do}{t} = \frac{6.625}{0.095} = 69.7$$

From Graph (Fig. G) in ASME Section II Part D,

Factor A = 0.000225

From Graph (Fig. HA-3) in ASME Section II Part D,

Factor B = 2750 for the above factor A and for 750^{0}F

Allowable pressure

$$Pa = \frac{4}{3} \frac{B}{Do/t}$$

$$= \frac{4 \times 2750}{3 \times 69.7} = 52.6 \text{ psig}$$

<u>This is less than the Design Pressure.</u>

Therefore, assume higher thickness.

Consider SCH 80 S pipe

Nominal thickness = 0.432"

Minimum thickness = 0.875 x 0.432

= 0.378"

$$\frac{Do}{t} = \frac{6.625}{0.378} = 17.5$$

Factor A for the new value of $\frac{Do}{t}$ is 0.0038

Corresponding factor B = 5500

Allowable Pressure;

$$Pa = \frac{4 \times 5500}{3 \times 17.5} = 419 \text{ psig}$$

<u>More than Design Pressure</u>

Hence select SCH 80S pipe.

1.7.3 THICKNESS OF BEND

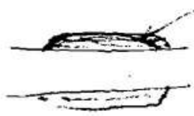
ASME B31.3, in it the 1999Edition, has added the formula as below for establishing the minimum thickness of bend.

The minimum thickness t_m of a bend

$$t = \frac{PD}{2\left(\frac{SEW}{I} + PY\right)}$$

after bending, in its finished form, shall be Where at the intrados (inside bend radius)

(thickness at extrados is more than intrados)



In case of jacketed pipe. The core is subjected to external pressure before internal pressure ... [illegible]

$$I = \frac{4\left(\frac{R}{D}\right)-1}{4\left(\frac{R}{D}\right)-2}$$

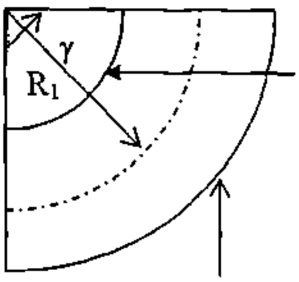
and at extrados

$$I = \frac{4\left(\frac{R_1}{D}\right)+1}{4\left(\frac{R_1}{D}\right)+2}$$

and at side wall the bend centre line radius

I = 1. The thickness apply at mid span $\gamma/2$.

Where W = Weld Joint strength reduction factor



Intrados

Extrados

Weld Joint Strength Reduction Factor 'W'

At elevated temperature, the long term strength of the weld joints may be lower than the long term strength of the bare material. For welded pipe, the product of the allowable stress and the applicable weld quality ~~factor~~ SE shall be multiplied by the weld joint strength reduction factor W when determining the required wall thickness.

The weld joint strength reduction factor is the ratio of the nominal stress to cause failure of the weld joint to that of the base material for the same duration in absence of the more applicable data the factor shall be taken as 1.0 at temperatures 510^0 C (950^0 F)

between 510 & 815, we have to interpolate.

above

and below and 0.5 at 815^0C ($1500^{0°}$ F) for all materials. This factor shall be linearly interpolated for intermediate temperatures.

after 815°C Et will be 0.5

2.0 PIPE FITTINGS

The branching tree shown (refer Fig.2.1) indicates the various types of fittings. These fittings can have various types of end connections or can have combination of end connections. The dimensional standards referred for the fittings are as follows:

DIMENSIONAL STANDARDS

1. ASME B 16.1
   - Cast Iron Pipe Flanges and Flanged Fittings
2. ASME B 16.3
   - Malleable-Iron Threaded Fittings
3. ASME B 16.4
   - Grey Iron Threaded fittings
4. ASME B 16.5
   - Pipe Flanges and Flanged Fittings
5. ASME B 16.9
   - Factory-Made Wrought Steel Butt welding
6. ASME B 16.11
   - Forged Fittings, Socket welding and Threaded
7. ASME B 16.28
   - Wrought Steel Butt welding Short Radius Elbows and Returns
8. ASME B 16.42
   - Ductile Iron Pipe Flanges and Flanged Fittings
9. ASME B 16.49
   - Buttwelding Induction Bends for Transportation and Distribution System
10. BS 1640
    - Butt weld Fittings
11. BS 3799
    - Socket weld and Screwed end fittings
12. BS 2598
    - Glass Pipelines and Fittings
13. IS 1239 Part-II - M.S. Fittings
14. IS 1538 - Cast Iron Fittings

15. MSS-SP-43
  - Stainless Steel Fittings

2.1 Classification Based On End Connections

2.1.1 SOCKET WELD/SCREWED END FITTINGS

For Socket Weld/Screwed end fittings are covered under ASME B 16.11/BS 3799. For these fittings, four pressure classes are available.

They are;

1 2000 # Class -
2 3000 # Class
3 6000 # Class
4 9000 # Class

These designations represent the maximum cold non-shock working pressure of the fitting in pounds per square inch.

1. 2000 # Class

This class is applicable only to screwed fittings and is covered only in ASME B 16.11. The corresponding pipe thickness for this class is SCH 80 or XS.

2. 3000 # Class

This class is applicable to both screwed and socket weld fittings. The corresponding pipe thickness for this class is SCH 80 or XS for socket weld end connection and SCH 160 for screwed end connections.

3. 6000 # Class

This class is also applicable to both screwed and socket weld fittings. The socket weld fittings under this class are normally used with SCH 160 pipes and screwed fittings with XXS pipes.

4. 9000 # Class

This class is applicable only to socket weld fittings, which are normally, used with XXS pipes.

The screwed end fittings can be with parallel threads or with taper threads. Taper threads are preferred for the fittings. These could be to NPT as covered in American Standards or to BSPT as covered in British standards or to relevant Indian Standard specifications.

The dimensional standard ASME B 16.11/BS 3799 cover the sizes upto 4" (100 mm) NB only.

The socket weld /screwed fittings are manufactured by forging. The materials of construction used for the same are as follows:

SW/SCRD FITTING MATERIALS

1 ASTM A105 - Forged Carbon Steel
2 ASTM A181 - Forged Carbon Steel for General Purposes
3 ASTM A182 - Forged Alloy Steel and Stainless Steel
4 ASTM A234 - Wrought Carbon Steel and Alloy Steel pipe fittings for moderate and elevated temperatures
(Similarly ASTM A105 base material is same)
5 ASTM A350 - Forged Alloy Steel for Low Temperature Services

2.1.2 BEVELED END FITTINGS

These types of fittings are connected by means of butt welding. The thickness of these fittings is to be specified the same as that of pipes because the bore of the pipes and the attached fittings should match. That means both the items should have the same schedule number. There are certain exceptional cases where fittings of higher thickness are used.

The beveled end fittings could be of seamless or welded construction.

The material of construction specified in the American Standards for the beveled weld fittings are;

BW FITTING MATERIALS

1. ASTM A 234 - Carbon Steel fittings for Moderate & High temperature Service
2. ASTM A 403 - Austenitic Stainless Steel Pipe fittings

S
W
WX
WU

Grade

specifi

A 234 GR.WPB W U

ASTM Specification

Weld

test
Ultrasonic

Carbon steel spec

A 403 GR.WP W U 304L

3. ASTM A 420 – Carbon Steel & Alloy Steel Pipe Fittings for low temperature services.

4. ASTM A 815 – Ferritic, Ferritic/Austenitic and Martensitic Steel Pipe Fittings

Beveled end fittings are covered under ASME B 16.9, B 16.28 and BS 1640.

Table of SCW/SW material

WCA → Carbon steel casting UTS – 40,000
WCB → UTS – 60,000
WCC → UTS – 70,000

SCW/SW material Rating	End Type	pipe based rating SCH no	Wall
2000	Threaded end	80	XS
3000	"	160	—
6000	"	—	XXS
8000	Socket weld	80	XS
6000	"	160	—
9000	"	—	XXS

In carbon steel two grades are there

① WPB – UTS – 60,000 (Wrought Steel product) WP

② WPC – UTS – 70,000

Wrought Steel Product

SS – 304 18/8 (Cr/Ni) – 0.08
304L – 0.035 (L = Low carbon steel)

SS – 316 18/8/3 (Cr/Ni/Mo)

Format – specifications – standard Grade Welded/seamless Test

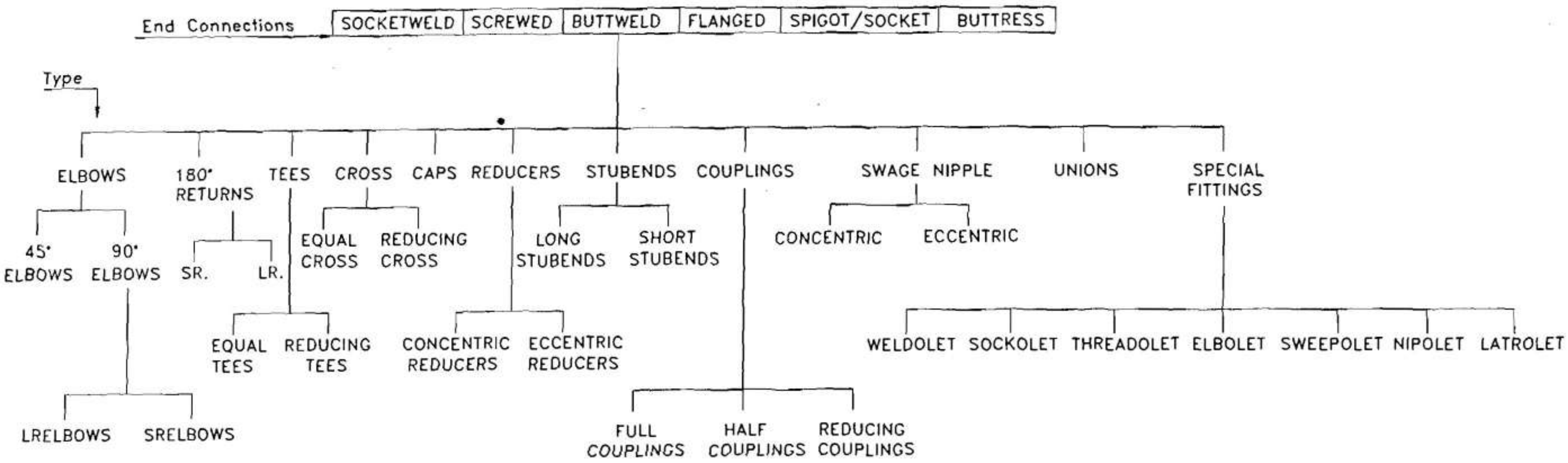


FIGURE 2.1

STANDARD PIPE FITTINGS

2.1.3 FLANGED END FITTINGS

Fittings with both ends flanged are used where welding is not possible or not permitted. Normally these are made by casting. Classification of these fittings, based on the pressure temperature ratings, is same as that of flanges.

Flanged fittings fabricated from standard butt-welded or socket welded fittings are not covered under this standard. The material specification is the same as that for castings.

FLANGED END FITTING MATERIALS

1. ASTM A 216 - Carbon Steel Castings
2. ASTM A 351 - Stainless Steel Castings
3. ASTM A 352 - Alloy Steel Castings
4. ASTM F 1545 - Plastic Lined Fittings
5. IS 1538 - CI Fittings

These fittings are covered under ASME B 16.5 and BS 1650 for carbon and alloy steel piping and ASME B 16.1 for cast iron fittings.

6. ASTM A 217 - High Temp alloy steel

2.1.4 SPIGOT SOCKET FITTINGS

Spigot Socket fittings are used in Cast Iron piping for low-pressure services. The joints are sealed by Lead caulking. This type of connection has the advantage that it can take misalignment to a certain extent. Flanged sockets and flanged spigots are used for connection to flanged equipments and valves. These fittings are covered under IS 1538.

2.1.5 BUTTRESS END FITTINGS

Buttress ends fittings are used in glass piping. These fittings are bolted together with the help of backing flanges and PTFE inserts. These fittings are covered under BS 2598.

2.2 Types Of Fittings

There are various types of fittings used to complete the piping system. These are used to change direction, change diameter or to branch off from main run of pipe. The special features of these are as below.

2.2.1 ELBOWS

Elbows are used to make 90 deg. or 45 deg. changes in the direction of run pipe. There are two types of 90 deg. butt-welding elbows available for use. These are the long radius and short radius elbows. The long radius elbows have a bend radius of 1.5D, where D is the nominal size, whereas the short radius elbows have a bend radius of 1D. The 45 deg. elbows are of 1.5D radius. Any bend with more than 1.5D bending radius has to be specially made as per requirements. For large diameter piping, bends are fabricated by profile cutting of pipes and are called mitre bends. Mitre bends with two piece, three piece or four piece construction can be made. These are normally not used in critical services. 22.5 deg. elbows are also available in cast iron construction.

SS	pipe	Forged	Fittings	Casting
304	A312 TP304	A182 Gr F-304	A403 Gr WP304	A351 Gr CF8
304L	A312 TP304L	A182 Gr F-304L	A403 Gr WP304L	A351 Gr CF3
316	A312 TP316	A182 Gr F-316	A403 Gr WP316	A351 Gr CF8M
316L	A312 TP316L	A182 Gr F-316L	A403 Gr WP316L	A351 Gr CF3M

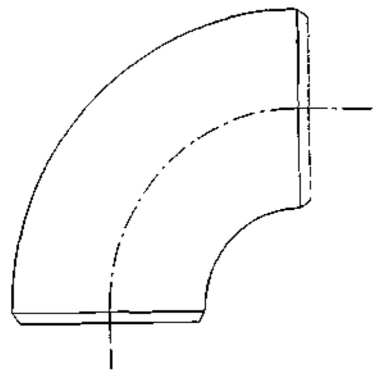


Fig. 2.2: Short Radius Elbow (R=1D)

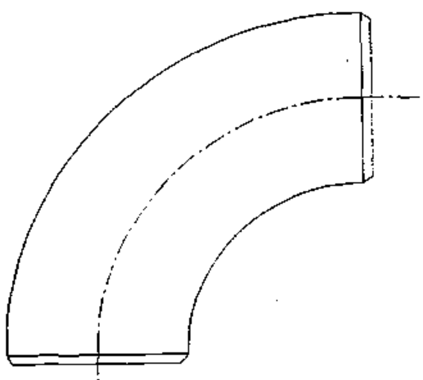


Fig. 2.3: Long Radius Elbow (R=1.5D)

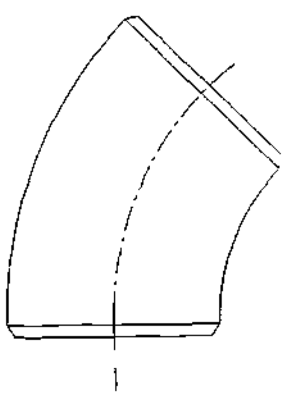


Fig. 2.4: Elbows - 45°

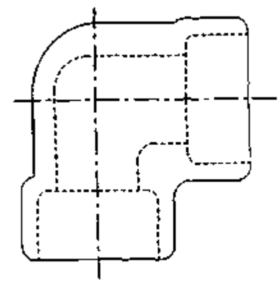


Fig. 2.5: Elbows – Socket weld

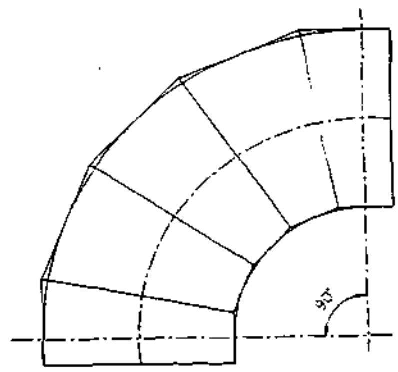


Fig. 2.6: Mitre Bend 90°

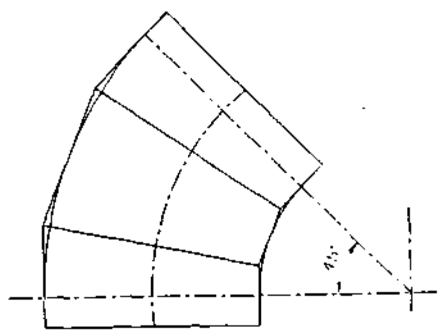
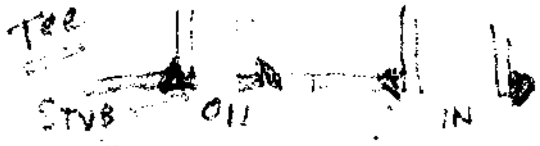


Fig. 2.7: Mitre Bend 45°



2.2.2 RETURNS

Returns change the direction through 180 deg. This is mainly used in heating coils, heat exchangers, etc. Returns with 1.5D radius and 1D radius are available.

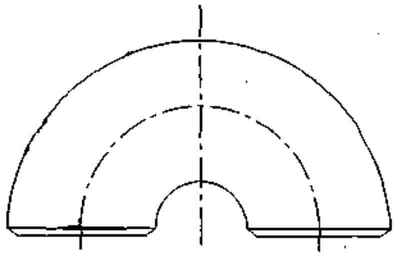


Fig. 2.8: Long Radius Return

2.2.3 TEES

Tees are used for branching off. For low pressure services, branching off is done by direct welding of branch pipe to run pipe instead of using a standard Tee. In certain cases, reinforcing pads are used for structural stability of such connections. Design code gives the calculation by which the requirement of reinforcement pad can be established and provided for branch connection (Refer Appendix H of ASME B 31.3). The branching schedule specified along with piping specification explains what sort of a branch connection is to be used for that particular piping class.

The manufacturing restrictions do not allow reducing tees of all size combinations. To arrive at available sizes of reducing tees in the standard, use the thumb rule of dividing the major diameter by 2 and consider the next lower size.

For example, the minimum size of reducing tee available for 4" NB size is 4" x 1½" (next lower size of 4/2 = 2").

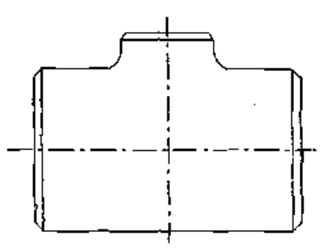


Fig. 2.9: Tees – Butt weld

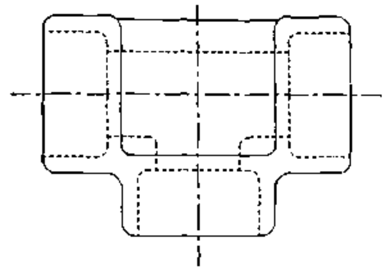


Fig. 2.10: Tees – Socket weld

2.2.4 CROSS

This is a fitting very rarely used in piping system. There are two types of crosses, the straight and reducing. To reduce the inventory, it is preferred to use tees except where space is restricted as in marine piping.

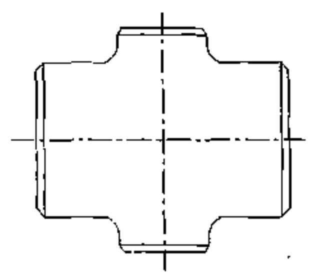


Fig. 2.11: Cross

2.2.5 REDUCERS

There are two types of reducers available, the concentric reducers and the Eccentric reducers.

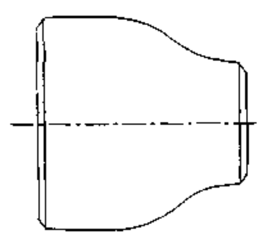


Fig. 2.12: Concentric Reducer

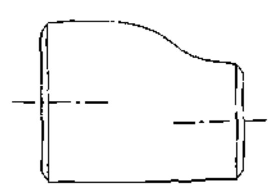


Fig. 2.13: Eccentric Reducer

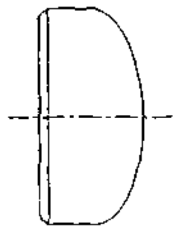


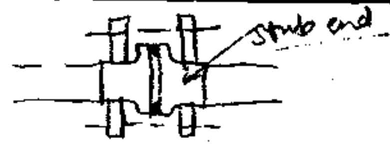
Fig. 2.14: Cap

When the center lines of the larger pipe and smaller pipe are to be maintained same, then concentric reducers are used. When one of the outside surfaces of the pipelines are to be maintained same, then eccentric reducers are required. There are no eccentric reducers in socket weld fitting and Swage nipples are used for such service. The size restrictions for manufacture as explained in Tees is also applicable to reducers.

2.2.6 STUB ENDS

To reduce the cost of piping, stub ends are used with backing flanges for flange joints when exotic materials are used in piping. ASME B16.9 specifies two types of stub ends, the long stub ends and the short stub ends. The length of stub ends as per MSS-SP-43 is the same as that of short stub ends. MSS-SP-43 specifies two classes, Class A with radius and Class B without radius at the corner. Class B can be used with slip-on flanges. Designer selects stub end (long/short) ensuring the weld of pipe to stub end not get covered by flange. When Class A stub ends are used, the inner diameter of backing flange is chamfered for better seating.

The minimum lap thickness should be the same as that of the pipe wall. When special facings such as tongue and groove, male and female etc. are employed additional lap thickness shall be provided. The gasket face finish shall be provided with serrations as required. ASME B 16.9



considered long pattern as the standard when nothing is specified in this respect.

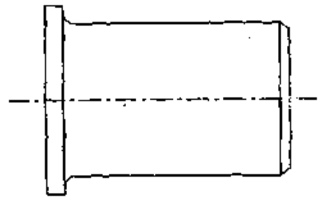


Fig. 2.15: Stub End – Class A

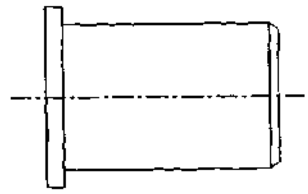


Fig. 2.16: Stub End – Class B

2.2.7 COUPLINGS

Couplings are of three types:

1.Full Coupling
2.Half Coupling
3.Reducing Coupling

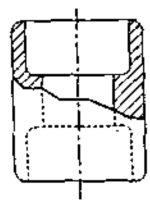


Fig. 2.17: Full Coupling

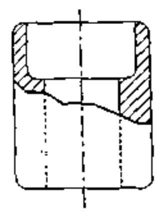


Fig. 2.18: Half – Coupling

Full couplings are used to connect small bore pipes as projection of welding inside the pipe bore, when butt welding is used, reduce the flow area. Half couplings are used for branch connections and reducing couplings for size reduction. Reducing

Eccentric Reducer ① TO maintain same level of support

For pump to avoid cavitation, essential

② Gas line

③ pump inlet/outlet

couplings maintain the pipe centerlines same and eccentric swage nipples are used to maintain the outside surface same for such systems.

2.2.8 SWAGE NIPPLES

Swage Nipples are like reducers but are used to connect butt welded pipe to smaller screwed or socket welded pipe. There are two types of swage nipples, the concentric and the eccentric. Various combinations of end connections are possible in swage nipples. These are designated as

PBE - Plain Both Ends
PLE - Plain Large End
PSE - Plain Small End
BLE - Beveled Large End
TSE - Threaded Small End

These are covered under the regulatory Code BS 3799.

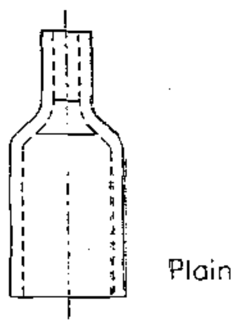


Fig. 2.19: Concentric Swage Nipple

Fig. 2.20: Eccentric Swage Nipple

2.2.9 UNIONS

Unions are used in low pressure piping where dismantling of the pipe is required more often, as an alternative to flanges. Unions can be with threaded end or with socket weld ends. There are three pieces in a union, two end pieces to attach to the run pipe and the third threaded piece to connect these two. The ball type metal seating ensure sealing.

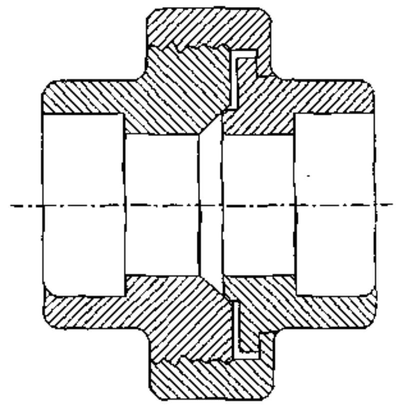


Fig. 2.21: Union

2.2.10 SPECIAL FITTINGS

The items referred under special fittings are;

* Weldolet
* Sockolet
* Threadolet
* Elbolet
* Sweepolet
* Nipolet
* Latrolet

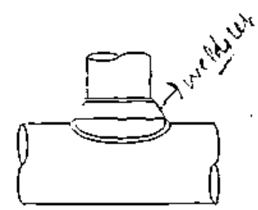


Fig. 2.22: Weldolet

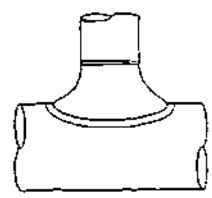


Fig. 2.23:Sweepolet

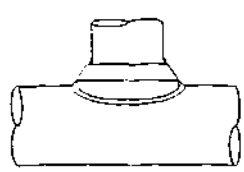


Fig. 2.24: Sockolet

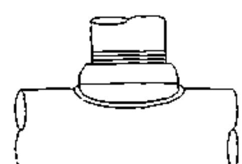


Fig. 2.25: Thredolet

(If thermowell is to be put on Elbolet, put against the direction of flow)

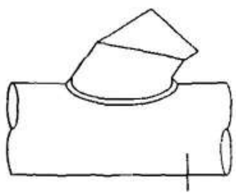


Fig. 2.26: Latrolet

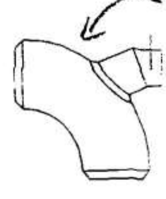


Fig. 2.27: Elbolet

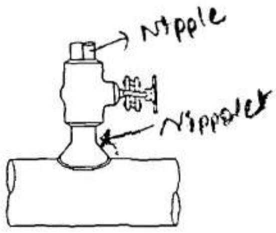


Fig. 2.28: Nipolet

These are fittings, which have restrictive use. Weldolet is used for butt-weld branch connection where standard tee is not available due to size restriction and the piping is of critical/high pressure service. Sockolet is used for socket welding branch connection, which require reinforcing pad. Threadolet is used for threaded branch connections. Elbolet is used for branch connection on elbows and have the profiles made to suit the elbow. Sweepolet is integrally reinforced butt weld branch connection. Latrolet is used for branch connection at an angle.

3.0 FLANGES

(AWWA → 62" to 90"
B16.5 → 1/2" to 24"
B16.47 → 26" to 60"

Flanges are used when the joint needs dismantling. These are used mainly at equipments, valves and specialties. In certain pipelines where maintenance is a regular feature, breakout flanges are provided at definite intervals on pipe lines. A flanged joint is composed of three separate and independent although interrelated components; the flanges, the gaskets and the bolting; which are assembled by yet another influence, the fitter. Special controls are required in the selection and application of all these elements to attain a joint, which has acceptable leak tightness. Classification of flanges is done in several alternate ways as follows:

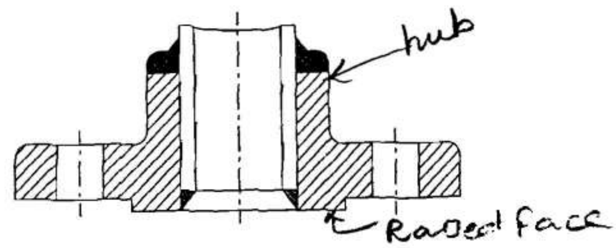


Fig. 3.1 : Slip-on Raised Face Flange

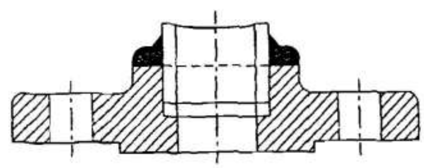


Fig3.2: Socket Welded Raised Face Flange

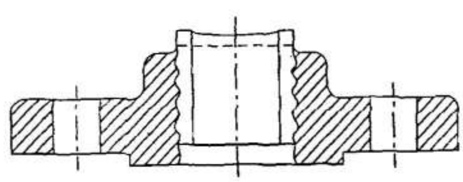


Fig. 3.3: Threaded Raised Face Flange

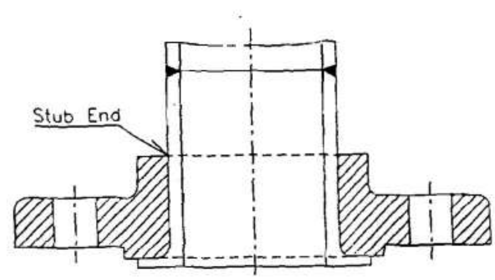


Fig. 3.4: Lap Joint Flange with Stub End

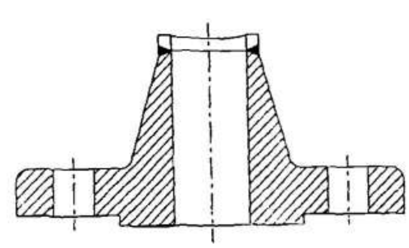


Fig.3.5: Welding Neck Raised Face Flange

In flanges, holes on flanges are off centre

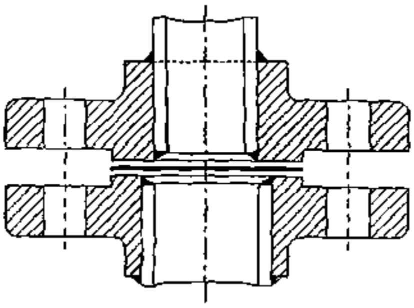


Fig.3.6: Reducing Slip-on Flange

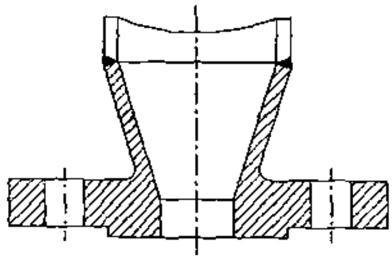


Fig.3.7: Expander or Reducer Flange

3.1 Based On Pipe Attachment

Flanges can be classified based on the attachment to the piping as below;

3.1.1 Slip-on
3.1.2 Socket Weld
3.1.3 Screwed
3.1.4 Lap Joint
3.1.5 Welding Neck
3.1.6 Blind
3.1.7 Reducing
3.1.8 Integral

The **Slip-on type** flanges are attached by welding inside as well as outside. Normally, these flanges are of forged construction and are provided with hub. Sometimes, these flanges are fabricated from plates and are not provided with the hub.

The **Socket weld flanges** are welded only on one side and are not recommended for severe services. These are used for small-bore lines only. The thickness of connecting pipe should be specified for this type of flanges to ensure proper bore dimension.

The **Screwed-on flanges** are used on pipe lines where welding cannot be carried out. Socket welding and threaded flanges are not recommended for service above 250°C and below -45°C.

The **Lap joint flanges** are used with stub ends when piping is of a costly material. The stub ends will be butt-welded to the piping and the flanges are kept loose over the same. The inside radius of these flanges is chamfered to clear the stub end radius. With Class B type stub ends slip-on flanges can be used for the same duty.

The **Welding neck flanges** are attached by butt-welding to the pipes. These are used mainly for critical services where all the weld joints need radiographic inspection. While specifying these flanges, the thickness of the welding end also should be specified along with flange specification.

The **Blind flanges** are used to close the ends, which need to be reopened later.

The **Reducing flanges** are used to connect between larger and smaller sizes without using a reducer. In case of reducing flanges, the thickness of the flange should be that of the higher diameter.

Integral flanges are those, which are cast along with the piping component or equipment. Thickness of integrally cast flanges and welded on flanges differ in certain sizes. There are some types of flanges developed by manufacturers, which are not covered in Code. They are mainly modification on the welding neck such as:

a) Long Welding neck flange

b) Expander/Reducer flange

Bolt holes are in multiples of four and shall straddle the fitting centerline.

3.2 Based On Pressure-temperature Rating

The flanges are also classified by the pressure temperature rating in ASME B 16.5 as below;

3.2.1	150 #
3.2.2	300 #
3.2.3	400 #
3.2.4	600 #
3.2.5	900 #
3.2.6	1500 #
3.2.7	2500 #

Pressure temperature rating charts, in the standard ASME B 16.5, specify the non-shock working gauge pressure to which the flange can be subjected to at a particular temperature. The indicated pressure class of 150#, 300#, etc. are the basic ratings and the flanges can withstand higher pressures at lower temperatures. ASME B 16.5 indicates the allowable pressures for various materials of construction vis - a -vis the temperature. ASME B16.5 does not recommend the use of 150# flanges above 400 °F (200 °C).

3.3 Based On Facing

The flanges can also be classified based on the facings as below:

3.3.1 Flat face (FF)
3.3.2 Raised face (RF)
3.3.3 Tongue and groove (T/G)
3.3.4 Male and Female (M/F)
3.3.5 Ring type joint (RTJ)

Flat face flanges are used when the counter flanges are flat face. This condition occurs mainly on connection to Cast Iron equipments, valves and specialties.

For 150# and 300# flanges, the raised face is of 1/16 inch and is excluded in the thickness specified. For higher rating, the flange thickness does not include the raised face thickness. The raised face thickness for higher rating is ¼ inch.

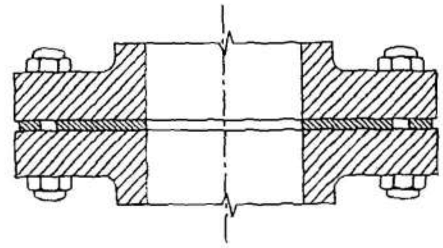


Fig. 3.8: Flat Face

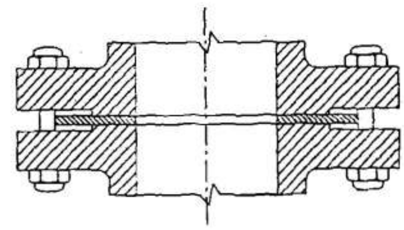


Fig. 3.9: Raised Face

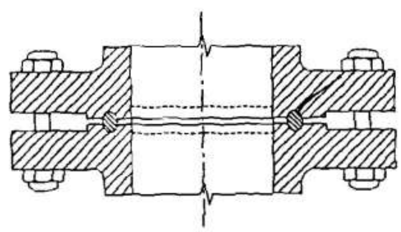


Fig. 3.10: Ring Joint

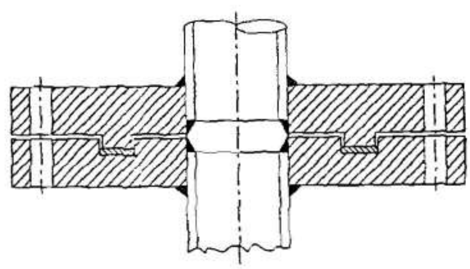


Fig. 3.11: Tongue and Groove Joint

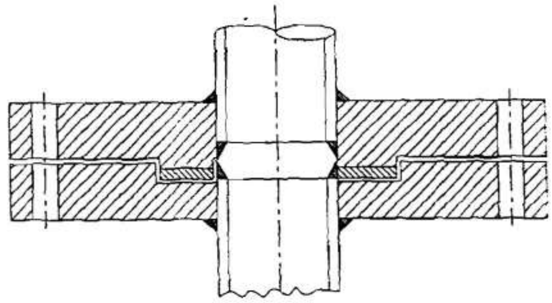


Fig. 3.12: Male / Female Joint

RF cannot be used in Al, CI application.

If any valve, equipment is ordered, it is ordered with groove and flange, so that it is not damaged.

3.4 Based On Face Finish

There are two types of finishes done on to the facings. They are the **smooth finish** and the **serrated finish**. The smooth finish flanges are specified when metallic gaskets are specified and serrated finish is provided when a non-metallic gasket is provided. The serrations provided on the facing could be concentric or spiral (phonographic). Concentric serrations are insisted for face finish when the fluid being carried has very low density and can find leakage path through the cavity. The serration is specified by the number, which is the **Arithmetic Average Roughness Height (AARH)**. This is the arithmetic average of the absolute values of measured profile height deviations taken within the sampling length and measured from the graphical centre line.

3.5 Based On Material Of Construction

The flanges are normally forged except in very few cases where they are fabricated from plates.

When plates are used for fabrication, they should be of weldable quality. ASME B16.5 allows only reducing flanges and blind flanges to be fabricated from plate. The materials of construction normally used are as follows;

FLANGE MATERIALS

3.5.1 ASTM A105	- Forged Carbon Steel
3.5.2 ASTM A181	- Forged Carbon Steel for General Purpose
3.5.3 ASTM A182	- Forged Alloy Steel and Stainless Steel
3.5.4 ASTM A350	- Forged Alloy Steel for low temperature services

3.6 Other Standards

Certain British Standards, German Standards and Indian Standards are also followed in India for flange specifications. BS-10 is the most popular among them. DIN flanges are also popular because they have a wider range of pressure temperature classes. IS has developed IS 6392 in line with DIN standards and the same is also in use.

ASME B 16.5 Covers Sizes from ½" NB to 24" NB only and ANSI B16.47 / API 605 are referred for higher sizes.

physical
- Resilience
- Impermeability
- Compressibility
- Stress Relaxation

4.0 GASKETS

4.1 Selection

Proper selection of gasket depends upon following factors.

4.1.1 Compatibility of the gasket material with the fluid.

4.1.2 Ability to withstand the pressure-temperature of the system.

4.2 Type

Based on the type of construction, gaskets are classified as:

- 4.2.1 Full Face
- 4.2.2 Inside bolt circle
- 4.2.3 Spiral wound metallic
- 4.2.4 Ring type
- 4.2.5 Metal jacketed

- Chemical Resistance
- Capacity to withstand pr., temp
- Anticorrosive
- Antistick

Properties

4.3 Material

Experience on the job and published literature shall be used to select the gasket material with respect to the compatibility of the same with the fluid.

The material, which is most commonly used, is the Compressed Asbestos Fibre.

Indian Standard IS 2712 specifies three different materials at three different grades.

4.3.1 IS 2712 Gr W/1, W/2 and W/3
- for Steam, Alkali and general applications.

4.3.2 IS 2712 Gr A/1,
- for Acid applications.

4.3.3 IS 2712 Gr O/1, O/2, O/3
- for Oil applications.

Asbestos free gaskets are also available for above applications. For very corrosive applications, PTFE or PTFE enveloped gaskets are used.

Gasket Material

(1) Non-metallic
- Paper, PTFE
- Cork
- Rubber bonded cork

(2) semi-metallic
- Metal clad Gasket
- Spiral wound Gasket

For high temperature and high-pressure applications, spiral wound metallic gaskets are used. The selection of material of construction for winding depends upon the corrosive nature and concentration of the fluid, the operating temperature and the relative cost of alternate winding materials. The most commonly used are the Austenitic stainless steel 304, 316 and 321 with Asbestos filler. For very high temperatures, graphite filler is also used. Alternate winding materials also can be used depending upon the services.

ASME B 16.5 does not recommend the use of 150# rating spiral wound gaskets on flanges other than welding neck and lapped joint type.

Spiral wound gaskets are provided with carbon steel external ring known as centering ring to position the gasket. When used in vacuum services, an internal ring is also provided. The material of inner ring should be compatible with the fluid. The spiral wound gasket will perform when the flange face is 125-250 AARH finish.

4.4 Dimensional Standards

Gasket dimensions are covered under the following standards.

4.4.1 API 601
- Metallic Gasket for Refinery Piping

4.4.2 BS 3381
- Metallic Spiral Wound Gaskets

4.4.3 ANSI B 16.20 (ASME B16.20)
- Metallic Gaskets for pipe flanges

4.4.4 ANSI B 16.21 (ASME B16.21)
- Non-metallic Gaskets for pipe flanges.

5.0 BOLTING

Depending upon the service, its pressure/temperature and the type of gasket, type of bolting is selected.

For low pressure, low temperature services, machined bolts are used and studs are used otherwise. Normally, the bolts are provided with hexagonal head, hexagonal nut and a round washer. Studs are provided with two hexagonal nuts and two washers. The length of bolts/studs required for the flange joints of all pressure classes are specified in ASME B16.5.

Flanged joints using low strength carbon steel bolts shall not be used above 200 °C or below –29 °C

ASTM F-704 specifies the standard practice of selecting bolt lengths for piping system-flanged joints.

5.1 Material Of Construction For Bolting

Bolting materials normally used are:

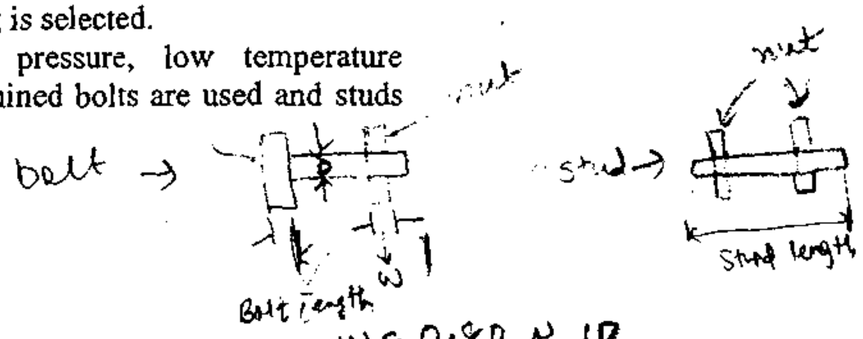
5.1.1 ASTM A 307	-Low Carbon Steel Bolting Material
5.1.2 ASTM A 320	-Alloy Steel Bolting material
5.1.3 ASTM A 563	- Carbon and alloy steel nuts
5.1.4 ASTM A193	- Alloy Steel Bolting Material for high temperature service
5.1.5 ASTM A 194	- Alloy Steel nut material for high temperature service
5.1.6 IS 1367	- Threaded steel fasteners

5.2 Dimensional Standards For Bolts

The dimensional standards referred for the studs/bolts are:

flanges

5.2.1 ANSI B 18.2.1	- Square & Hexagonal head bolts
5.2.2 ANSI B 18.2.2	- Square & Hexagonal nuts
5.2.3 BS 916	- Black bolts & nuts
5.2.4 IS 1367	- Threaded steel fasteners.



6.0 NON- FERROUS PIPING

The non-ferrous piping is used depending upon the corrosion properties and the temperature at which the fluid is handled. Special technology is involved in the fabrication of these piping. The commonly used materials are:

* Aluminum
* Alloy-20
* Hastalloy
* Lead
* Monel
* Nickel
* Titanium

These materials are specified under ASTM Section II part B and the numbers are prefixed with the Alphabet 'B'.

Due to economic considerations either carbon steel flanges with lining/bonding of these materials or Lap joint backing flanges wherever possible are used in this piping.

7.0 NON-METALLIC AND LINED PIPING

Non-metallic piping is used where the problem of corrosion is severe and it is difficult to get a suitable economical metallic piping. Temperature limitations restrict the use of these non-metallic piping. The commonly used materials are:

ABS – Acrylonitrile-Butadiene-Styrene
CPVC – Chlorinated Polyvinyl Chloride
ETFE – Ethylene Tetrafluoroethylene
FEP – Fluoro Ethylene propylene
FRP – Fibreglass Reinforced Plastic
HDPE – High Density Polyethylene
LDPE – Low Density Polyethylene
PFA – Perfluoro Alkoxyalkane
PP – Polypropylene
PTFE – Polytetrafluoroethylene
PVC – Poly Vinyl Chloride
PVDF – Polyvinyliedene Fluoride
Glass
Cement
Ceramic

To add mechanical strength with the corrosion properties of non-metallic materials, the concept of lining of material is established. The combination normally used in the industry are:

* Mild Steel Rubber Lined (MSRL),
* Mild Steel Glass Lined (MSGL),
* Mild Steel Cement Lined,
* Mild Steel PP Lined,
* Mild Steel PTFE lined
* Mild Steel PVDF lined

The lined pipes and pipe fittings have flanged ends and are joined by bolting. Of late flangeless lined piping is in use. In this case the liner is butt-welded and the outer carbon steel shell of the pipe is connected by 'Lorking' mechanical coupling.

The use of gasket is not recommended in piping lined with resilient materials, but this can damage the lining restricting the reuse.

The requirement of lined pipes has to be studied case by case based on the service conditions.

The glass pipes & fittings have either buttress end or beaded ends and are connected with flange assembly.

8.0 PIPING SPECIFICATION / PIPING CLASS

A document indicating the dimensional and material specifications of pipes, fittings and valve types is called a PIPING CLASS. Each class represents distinct features such as pressure-temperature conditions, corrosion resistance and strength abilities or a combination of these abilities. There could be a number of them selected and used for one project. While selecting these, care should be taken to minimize the number to rationalize the inventory. The designation of these Piping Classes varies with the company. While designing the piping system for a project, the components, which are not mentioned in the piping class, should be avoided.

9.0 TIPS FOR THE PREPARATION OF PIPING SPECIFICATIONS

The approach should be to minimize the number of different elements and thus simplify and rationalize inventory.

9.1 Materials

* Carbon Steel shall be used for temperature upto 425°C (800 °F) only.
* Low temperature steel shall be used for temperature below -29 °C (-20 °F)
* Alloy steel shall be used for temperature above 426 °C (801 °F)
* Stainless steel shall be used for corrosive fluids. Basic material of construction specified by Process Licenser to be referred for the type.
* Galvanized steel piping shall be used for services such as drinking water, instrument air, nitrogen (LP) etc.
* Selection of Non-ferrous, Non-metallic and Lined piping shall be as per the recommendation from the Process Licenser.

9.2 Piping Joints

* Butt-welded connection shall normally be used for all Alloy/Carbon steel piping 2" (50 mm) NB and larger and also for Austenitic Stainless Steel.
* Alloy/Carbon steel piping 1½" (40 mm) NB and below shall be socket welded.
* Threaded connection shall be avoided except in galvanized piping.
* Flanged joints shall be minimized, as they are points of potential leakage. It may be used to connect piping to equipment or valves, connecting pipe lines of dissimilar materials, where spool pieces are required to permit removal or servicing of equipment and where pipes and fittings are with flanged ends.

9.3 Piping Components

9.3.1 PIPES

* All pipelines carrying toxic/inflammable fluids shall be seamless. .
* Utility piping can be ERW or Seam welded.
* Steam pipelines shall preferably be seamless.

9.3.2 FITTINGS

* Fittings shall preferably be seamless.
* Butt weld fittings shall be used for pipe sizes 2" (50 mm) NB and above for all Alloy/Carbon steel piping.
* For stainless steel piping where thickness is less, all fittings could be butt-welding type.
* Welding tees shall be used for full size branch connections. For reduced branch sizes upto 2 steps less than run diameter, it can be fabricated. For smaller sizes half couplings shall be used. Full size unreinforced branch welding can be done where pressure temperature condition are mild.

9.3.3 FLANGES

* Rating shall be based on the pressure temperature conditions. However 150 lb flanges are not permitted beyond 200°C (400°F).
* Socket welding flanges may be used for all pressure ratings upto 1½" (40 mm) NB size except on lines subjected to severe cyclic conditions..
* Screwed flanges shall be used for galvanized steel/cast iron piping.
* Slip on flanges are used in 150 lb and 300 lb rating upto a maximum of 200°C. Welding neck flanges shall be used for higher pressure ratings.
* Raised face is used for flanges upto 600 lb rating. For flanges 900lb rating and above RTJ is recommended. Tongue and groove facing shall be used selectively.

* Depending on pressure and temperature, gasket shall be either CAF, spiral wound metallic for raised face flanges or selected based on the corrosive nature of the fluid.
* Use flat face flanges to mate with cast iron valves and equipments.
* Use Spiral wound gasket with inner ring for Vacuum service
* Low strength carbon steel bolting shall not be used above 200 °C and below –29 °C

****

PIPING MATERIAL SPECIFICATION	Job No. xxx	PIPING CLASS A 41
SERVICE: Process (Low Temperature)	Job Code. AAA	
	Client: YYY	
	w.p:51 kg/cm²g	
	Temp.°C Max.amb Min.-50	
BASE MATERIAL: Alloy CS / **RATING AND FACING:** ANSI B16.5 400# RF	CA: 6 mm	SPL. I C

Item	Size Range Low	Size Range High	Type	MATERIAL	DIMN. STD.	THK./ RATING	ENDS	REMARKS
PIPES	15	40	smls	ASTM A333 Gr. 1	B36.10	XXS	BE	
	50	100			B36.10	SCH 160	BE	100 % Radiography
	150	300			B36.10	SCH 80	BE	100 % Radiography
FLGS.	15	40	SW	A 350	B16.5	400#	RF	125 AAAH SERR. FINISH
	50	100	WN		B16.5	400#	RF	to suit pipe SCH. 160/80
	150	300	WN	Gr.LF2	B16.5	400#	RF	respectively
B.F.	15	300	Forged	A350 Gr. LF2	B16.5	400#	RF	SERRATED 125 AARH
FITTINGS	15	40	Forged	A350GrLF2	B16.11	9000#	SW	Use 5D Bends instead of Elbows. Use Swage Nipple for reducers.
	50	100	Smls	ASTM A420 Gr. WP6	B16.9	SCH160	BE	
	150	300	Smls			SCH 80	BE	
GSKT	15	300	S/W with CS C.R.	SS 304 With CAF filler	B16.20 ~~API 601~~	4.5 MM		Cent.ring 3 mm thk (C.R.) → centering ring
BOLT	15	300	Studs with	A 193 Gr. B7 A194 Gr. 2H	B1.1 B18.2.2	Fully threaded	-	With 2 Nos. washers
VALVES	15	300	Gate	A 217 Gr. LCB	API 600	400#	BE	PTFE impregnated asbs pkg.
	15	300	Globe		BS 1873	400#	RE	PTFE impregnated asbs pkg.
	15	40	Lift	A 350 Gr. LF2	BS 5352	1500#	SW	
	15	50	Swing	A 217 Gr. LCB	BS 1868	400#	BE	

BRANCH SCHEDULE

HEADER	15	20	25	40	50	80	100	150	200	250	300	350	400	450	500	600
15	1															
20	1	1														
25	1	1	1													
40	1	1	1	1												
50	5	5	5	4	4											
80	5	5	5	5	4	4										
100	5	5	5	5	4	4	4									
150	5	5	5	5	7	4	4	4								
200	5	5	5	5	7	4	4	4	4							
250	5	5	5	5	7	7	4	4	4	4						
300	5	5	5	5	7	7	7	4	4	4	4					
350																
400																
450																
500																
600																
BRANCH	15	20	25	40	50	80	100	150	200	250	300	350	400	450	500	600

KEY
1. SW TEE
2. SCRD TEE
3. SW / SCRD HALF COUPLING
4. BW TEE
5. SOCKOLET
6. THREDOLET
7. WELDOLET
8. REINFORCED SET-ON
9. UNREINFORCED SET-ON

PIPING MATERIAL SPECIFICATION	Job No. xxx	PIPING CLASS A 61
SERVICE: Process (High Temperature)	Job Code. AAA	
	Client:YYY	
	w.p: 60 kg/cm^{2}g	
	Temp.Max.600 °C Min.AMB	
BASIC MATERIAL: Alloy CS — RATING AND FACING: ANSI B16.5 600# RF	CA: 3 mm	SPL. I H

Item	Size Range Low	Size Range High	Type	MATERIAL	DIMN STD.	THK./ RATING	ENDS	Remarks
PIPES	15	40	Smls	ASTM A335 Gr. P9	B36.10	SCH 160	PE	
	50	50				SCH 160	BE	10% Radiography
	80	150				SCH 80	BE	
FLGS.	15	40	SW	A 182 Gr. F 9	B16.5	600#	RF	125 AARH serr. finish & to suit SCH 160 / 80 pipe. respectively
	50	50	WN					
	80	150	WN					
B.F.	15	150	Forged	A 182 GR. F9	B16.5	600#	RF	125 AARH serr. finish
FITTINGS	15	40	Forged	A 182 GR. F9	B16.11	6000#	SW	Use swage nipples for reducers
	50	50	Smls	A 234 Gr. WP9	B16.9	SCH 160	BE	
	80	150	Smls	A234 Gr. WP9	B16.9	SCH 80	BE	
GSKT	15	150	S/W with CS(C.R)	SS 304 with CAF filler	~~API 601~~ B 16.2	4.5 mm		Cent. Ring 3 mm thk. (C.R.) → Centering Ring.
BOLT	15	150	Studs with	A 193 Gr. B7 / A 194 Gr. 2H	B1.1 / B18.2.2	Fully threaded		With 2 Nos. washers
VALVES	15	40	Gate	A 182 Gr. F9	BS 5352	800#	SW	Stellited trim.
	50	150	Gate	A 217 Gr WC12	API 600	600#	Flanged	Stellited trim.
	15	40	Globe	A 182 Gr. F9	BS 5352	800#	SW	Stellited trim.
	50	150	Globe	A 217 Gr. WC12	BS 1873	600#	Falnged	Stellited trim.
	15	40	Lift	A 182 Gr. F9	BS 5352	800#	SW	Stellited trim.
	50	150	Swing	A 217 Gr. WC12	BS 1868	600#	Flanged	Stellited trim.

BRANCH SCHEDULE

HEADER \ BRANCH	15	20	25	40	50	80	100	150	200	250	300	350	400	450	500	600
15	1															
20	1	1														
25	1	1	1													
40	1	1	1	1												
50	3	3	3	4	4											
80	3	3	3	3	4	4										
100	3	3	3	3	9	4	4									
150	3	3	3	3	9	9	4	4								
200																
250																
300																
350																
400																
450																
500																
600																

KEY
1. SW TEE
2. SCRD TEE
3. SW / SCRD HALF COUPLING
4. BW TEE
5. SOCKOLET
6. THREDOLET
7. WELDOLET
8. REINFORCED SET-ON
9. UNREINFORCED SET-ON

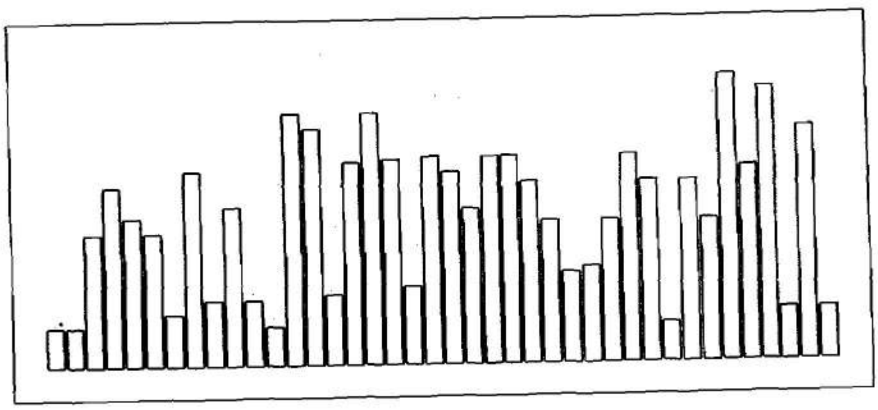
Certificate Course
on

PIPING ENGINEERING

June 12 - 25, 2006

VALVES

T. N. Gopinath
Consultant



Organized by

Piping Engineering Cell
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Powai, Mumbai - 400 076

VALVES

T. N. GOPINATH

INTRODUCTION

Estimates reveal that a substantial portion, approximately 8-10%, of the total capital expenditure of the chemical process industry is used for the procurement of valves. In terms of the number of units also, valves exceed any other piping component. Hence, proper thought should be given for the selection of valves. The first step in the selection is to determine exactly what function the valve is expected to perform after it has been installed.

Valves are installed on equipment/piping to perform any one of the following functions;

Functions of Valves

- Isolation
- Regulation
- Non-Return
- Special Purpose

The design of the valves are done in such a way as to perform any of the above functions. The type of valves used can be classified in the following categories.

1.0 ISOLATION

1.1 Gate Valves

1.2 Ball Valves

1.3 Plug Valves

1.4 Piston Valves

1.5 Diaphragm Valves

1.6 Butterfly Valves

1.7 Pinch Valves

2.0 REGULATION

2.1 Globe Valves

2.2 Needle Valves

2.3 Butterfly Valves

2.4 Diaphragm Valves

2.5 Piston Valves

2.6 Pinch Valves

3.0 NON-RETURN

3.1 Check Valves

4.0 SPECIAL PURPOSE

4.1 Multi-port Valves

4.2 Flush Bottom Valves

4.3 Float Valves

4.4 Foot Valves

4.5 Line Blind Valves

4.6 Knife Gate Valves

The above classification is based on functions. The valves could also be classified based on the type of construction. Valve manufacturers offer endless varieties of constructions. Based on the operation, valves can be broadly classified as operated valves and self-operated valves. Mainly the check valves are self-operated and all other types come under operated valves.
The valves can further be classified based on the end connections. End connection means the arrangement of attachment of the valves to the equipment or to the piping. The types of end connections are:

1.0 Screwed ends

2.0 Socket weld ends

3.0 Flanged ends

4.0 Butt weld ends

5.0 Socketted ends

6.0 Wafer type ends

7.0 Buttress ends

The valves could also be classified based on the materials of construction. There can be any number of combinations possible with the materials of construction. It is for the piping engineer to select the same in consultation with the process engineer to suit the process fluid. The environment in which the valves are installed is also to be considered for selection of materials of construction. However, the most commonly available materials are:

1.0 Cast Iron

2.0 Ductile Iron

3.0 Bronze

4.0 Gun metal

5.0 Carbon Steel

6.0 Stainless Steel

7.0 Alloy Carbon Steel ultra high

8.0 Poly Propylene, UHMW-PE, UHMW-HDPE etc.

9.0 Special Alloys

10.0 Fluoro polymer/Elastomer lined metals

11.0 Glass

TERMS USED FOR VALVES SPECIFICATION

1. Pressure - Temperature Ratings

Pressure - Temperature Rating is the maximum allowable sustained non-shock pressure at the corresponding tabulated temperature. These are listed in ANSI B 16.34 and ANSI B 16.5.

2. Class

The valve is specified by the pressure rating of the body of the valves. The American standard specifies the following classes.

2.1	Class	150#
2.2	Class	300#
2.3	Class	400#
2.4	Class	600#
2.5	Class	900#
2.6	Class	1500#
2.7	Class	2500#
2.8	Class	800# (SW
2.9	Class	4500#

3. Trim

The trim is comprised of Stem, Seat Surfaces, Back Seat Bushing and other small internal parts that normally contact the surface fluid. The table below indicates trim of common types of valves. API 600 specifies Trim numbers in table 3 of the standard. It specifies the types of material, which can be used for the parts with its typical specification and grade.

→ given in next pages

Trim Parts on common Valves

Gate	Globe	Swing Check	Lift Check
Stem ~~Seat Ring~~ Wedge Ring Bushing	Stem Seat Ring Disc nut Bushing	Seat Ring Disc holder Side plug Holder pin Disc nut pin	Disc guide Seat ring

4. Wetted Parts

All parts, which come in contact with the service fluid, are called the wetted parts.

5. Wire Drawing

This term is used to indicate the premature erosion of the valve seat caused by excessive velocity between seat and seat disc. An erosion pattern is left as if a wire had been drawn between the seat surfaces. Excessive velocity can occur when the valve is not closed tightly. A WOG (Water-Oil-Gas, relatively cool liquids) disc is the best defense against wiredrawing because its resiliency makes it easier to close tightly. Discs of harder material are to be closed carefully to prevent wire drawing. In LPG Service, the wire drawing effect causes a threat of anti-refrigeration. The ice formation on the wedge will obstruct movement thereby increasing the leak through seat further.

6. Straight - Through Flow

This refers to the valve in which the closing element is retracted entirely so that there is no restriction of flow.

(e.g. (diaphram valves)

7. Quarter - Turn Valves

This refers to the valve where the entire operation of valve is achieved by 90 degrees turn of the closing element.

(e.g. butterfly valve)

8. Pressure Drop

Pressure drop is the loss of pressure through resistance across the valve while flows and is expressed in terms of equivalent length in pipe diameters.

Type of Valve	Position	Equivalent length in pipe dia (L/D)
Gate	Fully open	13
Globe	"	340
Angle globe	"	145
Swing check	"	50
Plug – Rectangular Plug Port	"	18
Ball – Regular port	"	40
Ball – Full port	"	8

[From Crane Catalogue 410

9. Upstream Pressure

This is the pressure of the fluid that enters the valve. This is sometimes referred to as inlet or supply pressure.

Stellited – sprayed with hard spray on the surface for

10. Downstream Pressure
This is the pressure of the fluid that is discharged from the valve. This is sometimes referred to as outlet or reduced pressure.

11. LDAR
Signifies "Leak Detection And Repair" to ensure that the fugitive emissions standards of EPA are met. Fugitive emissions are the minute amount of process media that escape into the atmosphere though gland packing along valve stem.

12. LAER
Signifies "Lowest Achievable Emission Rate". It is the minimum rate of fugitive emission, which is achieved by deploying proper sealing arrangement.

13. LEAKAGE CLASS

Leakage class	**Maximum seat leakage**
Class I	A modification of any class II, III or IV valve where design intent is the same as the basic class, but by agreement between user and supplier. No test is required.
Class II	0.5% of rated valve capacity
Class III	0.1% of rated valve capacity
Class IV	0.01% of rated valve capacity
Class V	5×10^{-4} ml per minute water per inch of orifice diameter per psi differential
Class VI	as per table below

Size, NB	ml. per min.	Bubble per minute
1	0.15	1
1.5	0.30	2
2	0.45	3
2.5	0.60	4
3	0.90	6
4	1.70	11
6	4.00	27
8	6.35	45

TRIM

Table 3- Nominal Seating Surface Materials

Trim No.	Nominal Trim	Seat hardness (HB[a], minimum)	Material Type	Typical Specification (Grade) Cast	Forged	Welded
1	F6	b	13 Cr	ASTM A217 (CA15)	ASTM A182 (F6a)	AWS A5.9 ER410
2	304	c	18-8 Cr-Ni	ASTM A351 (CF8)	ASTM A182 (F304)	AWS A5.9 ER308
3	F310	c	25-20 Cr-Ni		ASTM A182 (F310)	AWS A5.9 ER310
4	Hard F6	750[d]	Hard 13 Cr			
5	Hardfaced	350[d]	Co Cr-A[h]			AWS A5.13 ERCoCr-A
5A	Hardfaced	350[d]	Ni-Cr			
6	F6 and	250[d]	13 Cr	ASTM A217 (CA15)	ASTM A182 (F6a)	AWS A5.9 ER410
	Cu-Ni	175[f]	Cu-Ni			
7	F6 and	300[d]	13 Cr	ASTM A217 (CA15)	ASTM A182 (F6a)	AWS A5.9 ER410
	Hard F6	750[d]	Hard 13 Cr			
8	F6 and	300[d]	13 Cr	ASTM A217 (CA15)	ASTM A182 (F6a)	AWS A5.9 ER410
	Hardfaced	350[d]	Co Cr-A[h]			AWS A5.13 ER CoCr-A
8A	F6 and	300[d]	13 Cr	ASTM A217 (CA15)	ASTM A182 (F6a)	AWS A5.9 ER410
	Hardfaced	350[d]	Ni-Cr			
9	Monel	c	Ni-Cu alloy	—	Manufacturer's standard	—
10	316	c	18-8 Cr-Ni	ASTM A351 (CF8M)	ASTM A182 (F316)	AWS A5.9 ER316
11	Monel	c	Ni-Cu alloy	—	Manufacturer's standard	—
	Hardfaced	350[d]	Trim 5 or 5A	—		See Trim 5 or 5A
12	316 and	c	18-8 Cr-Ni	ASTM A351 (CF8M)	----	AWS A5.9 ER316
	Hardfaced	350[d]	Trim 5 or 5A		ASTM A182 (F316)	See Trim 5 or 5A
13	Alloy 20	e	19-29 Cr-Ni	ASTM A351 (CN7M)	----	AWS A5.9 ER320
14	Alloy 20	c	19-29 Cr-Ni	ASTM A351 (CN7M)	ASTM B473	AWS A5.9 ER320
	and			—	ASTM B473	
	Hardfaced	350[d]	Trim 5 or 5A		-----	See Trim 5 or 5A

a HB is Brinell hardness number symbol per ASTM E10 (formerly BHN)

b Body and gate seats 250 HB minimum, with 50 HB minimum differential between body and gate seats.

c Manufacture's standards hardness

d Differential hardness between body and gate seat surfaces not required.

e case hardened by nitriding to thickness of 0.005- inch (0.13 millimeters) minimum.

f Hardness differential between body gate seats shall be as per the manufacturer's standard.

g Manufacturer's standard, within 30 Ni minimum

h This classification includes such trademarked materials such as stellite 6, Stoody 6, and Wallex 6.

i Manufacturer's standard hardfacing, with 25 percent Fe maximum.

RECOMMENDATION FOR VALVES

CHEMICAL/BULK DRUG PLANT

SR.	SERVICE	Size NB	Isolation	Regulation	Non-return	Remarks
1	Steam H.P.	½" –1 ½"	a) C.S. Body stellited trim 800# Globe with SW ends b) C.S. Body stellited trim 800# Piston with SW ends c) C S Body SS Ball. Special PTFE seats. 800# SW ball valve	a) C.S. Body stellited trim 800# Globe with SW ends b) C.S. Body stellited trim 800# Piston with SW ends	C.S. Body Stellited trim 800# Lift check with SW Ends	Piston valves are costlier. From the point of view of Energy conserva-tion they are O.K.
		2" – 12"	a) CS Body Stellited trim 300#/150# Flgd Gate with flex Wedge b) CS Body 13% Cr trim 300#/150# Flgd Piston	a) CS Body stellited trim 300# /150# Flgd Globe b) CS Body 13% Cr trim 300#/150# Flgd Piston	a) CS Body stellited trim 300#/150# Flgd swing check	ALL VALVES TO BE APPROV-ED BY IBR
2.	Steam L.P.	½" – 1 ½"	a) CS Body 13% Cr. Trim 800# Globe with SW ends b) CS Body 13% Cr. Trim 800# Piston With SW ends c) C S Body SS Ball. Special PTFE seats. 800# SW ball valve	a) CS Body 13% Cr. Trim 800# Globe with SW ends b) CS Body 13% Cr. Trim 800# Piston With SW ends	C.S. Body 13% Cr trim 800# Lift check With SW Ends	No IBR APPR REQUD. FOR PR. < 3.5 Kg/cm^2 g.
		2" – 12"	a) CS Body a 13% Cr. Trim Flex wedge 150# Flgd Gate b) CS Body 13% Cr trim 150# Flgd Piston	a) CS Body 13%Cr trim 150# Flgd Globe b) CS Body 13% Cr trim 150# Flgd Piston	C.S. Body 13% Crtrim 150# Lift check With SW Ends	

	Service	Size (NB)	Isolation	Regulation	NRV	Remarks
3.	Condensate	½" – 1 ½"	a) C.S. Body SS ball PTFE seat 800# SW Ball b) C.S. Body 13% Cr. Trim 800# SW Globe	a) C.S. Body 13% Cr. Trim 800# SW Globe valve		
		2" – 12"	SAME AS LP STEAM			
4.	Utilities like Water, Air, LSHS Low sulphur heavy stock	½" – 1 ½"	a) C.S. Body SS Ball PTFE Seat 800# Scrd. Ball b) G.M. Body Bronze trim Scrd Gate to IS 778	a) C.S. Body 13% Cr. Trim 800# SW Globe b) G.M. Body Bronze Trim Scrd Globe to IS 778	a) C.S. Body 13% Cr. Trim 800# Lift check With SW Ends b) G.M. Body Bronze Trim Scrd check to IS 778	Bronze body not Recommended in acidic atmosphere
		2" – 12"	a) CI Body 13% Cr disc 125# wafer type Butterfly Gear Operator above 6" NB b) CI Body 13% Cr or 18% Cr trim 125# Flgd Gate to IS 14846 c) CS Body 13% Cr trim 150# Flgd Gate d) Cast Iron Body 13% Cr plug 125 # Fldg Lub. Plug Valve	a) CI Body 13% Cr disc 125# wafer type Butterfly Gear Operator above 6" NB b) CI Body 13% or 18% Cr trim 125# Flgd Globe c) CS Body 13% Cr trim 150# flgd Globe -------	a) CS Body 13% Cr. trim water type check b) CI Body 13% or 18% Cr tri 125# flgd swing check to IS 5312 c) CS Body 13 % Cr trim 150# flgd swing check -------	Recommendation for Air CS Body Ball or Gate for Isolation

	Service	Size	Isolation	Regulation	NRV	Remarks
5)	Hot Oil / Heating Fluid	½" – 1 ½"	a) C.S. Body Stellited trim Graphoil pkg 800# SW Globe b) C.S. Body 13% trim 800# SW Piston with suitable sealing rings	a) CS. Body Stellited trim Graphoil pkg 800# SW Globe b) CS. Body 13% trim 800# SW Piston with suitable sealing rings	a) C.S. Body stellited trim 800# SW lift check	
		2" – 12"	a) CS Body stellited trim Graphoil pkg 300# (Min) Flgd Gate with 125 – 250 AARH Flgd Finish b) CS Body 13% Cr trim Graphoil pkg 300 # (Min) Flgd Piston with Suitable sealing ring and 125 – 250 AARH Flgd finish	a) CS Body stellited trim graphoil pkg 300# (Min) Flgd Gate with 125 – 250 AARH Flgd finish b) CS Body 13% Cr trim Graphoil pkg 300 # (Min) Flgd Piston with Suitable sealing ring and 125 – 250 AARH Flgd finish	CS Body stellited trim 300# (Min) Flgd swing check with 125 – 250 AARH Flgd Finish	
6)	Chlorine (Dry)	½" – 12"	Ball valve with CS body Monel/ Hastalloy C ball & stem	Globe Valve with CS Body Monel / Hastalloy C trim	Check (Lift / Swing) valve CS Monel / Hastalloy C trim	
7)	Solvent/ Process (Carbon Steel)	½" – 12"	a) C.S. Body SS Ball 150 Flgd full port Ball valve with PTFE/ GFT seats b) CS Body 13% Cr plug 150# Flgd sleeved plug seats	a) C.S. Body 13% Cr. trim 150# Flgd Globe	a) C.S. Body 13% Cr. trim Flgd Lift check ½" to 1 ½" and Flgd swing check 2" & above	

	Service	Size	Isolation	Regulation	NRV	Remarks
	Solvent/ Process (Stainless Steel)	½" – 12"	a) SS Body SS ball 150 # Fldg full port ball valve with PTFE / GFT seats b) SS Body SS Plug 150# Flgd sleeved plug	a) SS body SS trim 150# Flgd Globe	SS Body SS trim 150# Flgd swing check	
	Solvent/ Process (Highly Corrosive)	½" – 12"	a) Ductile Iron body & plug lined with fluoropoymer plug valve b) Ball valve with suitable plastic body and ball	—	a) Ductile Iron body lined with fluoro polymer ball check valve b) ---do---	Suitability with Temperature to be checked

1.0 ISOLATION VALVES

The isolation valves used in the Process Industry are:

1.1 Gate Valves

1.2 Ball Valves

1.3 Plug Valves

1.4 Piston Valves

1.5 Diaphragm Valves

1.6 Butterfly Valves

1.7 Pinch Valves

Of these, the Butterfly, Diaphragm and Piston Valves can be used for regulating the flow as well. Similarly, the Globe Valve design could be modified to use it for positive shut-off purposes. The present trend in industry is to go for quarter turn valves for this duty due to ease of operation. The types of valves in this category are the Ball valves, Plug valves and Butterfly valves. Ball and Plug valves are also can be used for flow control with shaped port of the closing element. More over the design of quarter turn valves are inherently better suited for emission control applications. The linear stem movement of the gate and globe valve tends to open the path of emissions release and in its dynamic mode, emissions can be "dragged" along the stem.

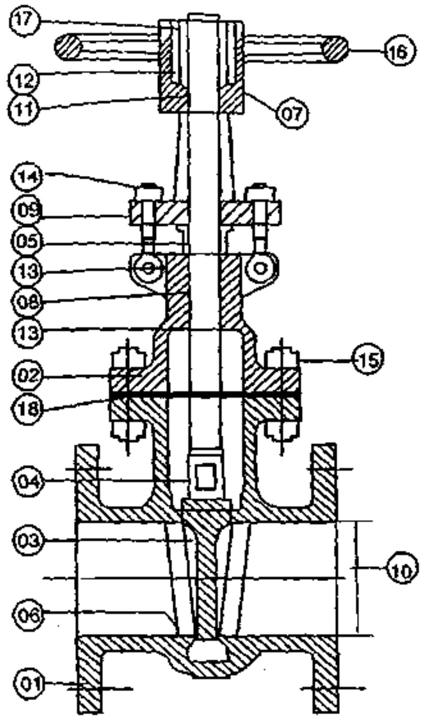
1.1. GATE VALVES

A typical Gate valve will have the following parts, which could be identified.

1.1.1 Body

1.1.2 Bonnet

1.1.3 Wedge



1.1.4 Stem

1.1.5 Gland

1.1.6 Seat ring

1.1.7 Yoke

1.1.8 Packing

1.1.9 Gland Flange

1.1.10 Valve Port

1.1.11 Yoke Bush

1.1.12 Lantern

1.1.13 Back Seat Bushing

1.1.14 Gland eyebolts & nuts

1.1.15 Bonnet bolts & nuts

1.1.16 Hand Wheel

1.1.17 Hand Wheel nut

1.1.18 Bonnet Gasket

1.1.1 Body

The body is the part which gets attached to the vessel or piping. The classification of the body could be done depending on the end connections as indicated earlier. Body could also be specified based on the material of construction of the same. This could be cast, forged or fabricated.

The wall thickness and end to end/face to face dimensions of the body shall be as per the Regulatory code to which it is designed.

The end flanges shall be integrally cast or forged with the body. It can also be attached by welding, if so specified. The end connection shall suit the rating specified. The flanged connection shall be to ANSI B 16.5 or any of the flange standards. The butt-welding end connection shall be to ANSI B 16.25 or any other end preparation required. The socket weld/screw connection shall be to ANSI B 16.11 or any other equivalent standards. The body can have auxiliary connection such as drains, by-pass connections, etc.

1.1.2 Bonnet

The bonnet is classified based on the attachment of the same to the body. The type of connection normally adopted are Bolted, Bellow sealed, Screwed-on, Welded, Union, Pressure sealed etc.

The bolted connection shall be flanged, male and female, tongue and groove or ring type joint. In low pressure rating valves, it may be flat faced. The bonnet gasket is selected to suit body-bonnet connection. It can be corrugated flat solid metal, flat metal jacketed, asbestos filled, metal ring joint, spiral wound asbestos filled or flat ring compressed asbestos in case of low pressure rating, Teflon or Teflon filled for corrosive applications.

The bellow sealed bonnets can be bolted or welded on to the body. These are selected for very critical services like the nuclear, very high temperature and lethal services. The screwed-on bonnet/union bonnet is used for very low priority application and small size valves.

When valves are used for Cryogenic Service extended bonnet design is used to take care of large insulation thickness. When used for very high temperature bonnet attached with fins are also used.

1.1.3 Wedge

This is the part, which facilitates the service by its movement up and down. The types of wedges are classified as;

- Solid Plain Wedge
- Solid Flexible Wedge
- Split Wedge

When solid disc is wedged into the rigid body seat and the valve undergoes temperature changes, the wedge gets jammed in the seat. Hence the flexible wedge and split wedge design is developed to overcome this difficulty.

Normally the solid plain wedge is referred as solid wedge and the split wedge is referred as flexible wedge. The design slightly alters with the manufacturers though the basis remains the same.

The flexible wedge design is followed for valve sizes 50 NB and above. Valves 40 NB and below are available in solid wedge design only. Flexible wedge design is superior as it will not get jammed during high temperature operations.

The wedge material should be at least of the same quality as that of the body. In case of integral seat rings the wedge circumference is deposited with superior quality material. In smaller valves, the whole wedge will be manufactured out of superior material.

Reduced bore - Inlet of valve - same dia. of pipe + size of valve - dia. of seat ring is less.

Pressure sealed bonneted valves are used in class country pipe lines. In these valves

1.1.4 Stem

The stem connects the hand wheel and the wedge for operations. The design can have rising stem and non-rising stem. The stem is operated rotating the stem nut by hand wheel mounted at the top of the yoke.

In the rising stem design, the stem moves up along with the wedge to open. This is called the OS & Y (Outside Screw and Yoke) type of design. In case of non-rising stem the wedge moves up and down and the stem is stationary. This is called the inside screw non rising stem design.

Normally, bar stock or forging are used for the construction of stem.

1.1.5 Gland, Gland flange, Packing and Lantern

There are two types of gland designs possible, Single piece and Two piece. In two-piece design, there will be gland flange and a follower. The follower will have a spherical end, which facilitates proper aligning of follower and loading on the packing. In Single piece, the gland and follower will be integral. This design is used mostly in low-pressure valves.

Normally gland follower will be of superior material than the gland flange. Gland flanges are made of carbon steel only. The glands are bolted to the bonnet with gland eyebolts in low-pressure valves.

The regulatory codes specify that the stuffing box should accommodate minimum six packing rings for class 150 valves. As regards higher rating valves, it should have lantern ring with five packing rings above and two packing rings below lantern. Lantern is not provided for class 150 valves. Lantern is provided for higher rating if required. When lantern is provided, the stuffing box shall be provided with two plugged holes. The material of lantern shall have corrosion resistance equal to that of the body.

Normally, the packing is of braided asbestos with suitable corrosion inhibitor. When special packing such as 'Graphoil' is used, the number of packing rings required will be more. To accommodate more packing rings, the length of gland is also modified. This design is called the 'Deep Gland' design. This is used for the high temperature services. But this cannot satisfy the EPA's fugitive emission standard of <500 PPM threshold. Hence frequent LDAR will result in excessive expenditure.

1.1.6 Seat Rings

There are two types of designs possible in seat rings. They are the integral and renewable. In case of renewable seat rings, it may be either threaded, rolled-in or welded-in. In case of integral seat rings, the seat material is weld-deposited directly on to the valve body. The minimum hardness specified by the code for this material is 250 HB, with 50 HB minimum differential between body and gate seats, the body seat being harder. Deposition of harder materials like "Stellite-6" is also done for valves used in special services.

The back seat arrangement is provided to repack the stuffing box when the gate is in fully open position. The stem shall have an integral conical or spherical backseat surface to seat against the bonnet backseat.

1.1.7 Yoke and Yoke Bush

Yoke may be integral with or separate from the bonnet. When the yoke is integral, the stem nut should be removable without removing bonnet. The yoke should have the same material of construction as that of the shell. The Yoke bush is normally a Ni-resist material. This is to prevent gauling of the stem, as stem will normally be of a Nickel alloy.

1.1.8 Hand wheel and Hand wheel Nut

The hand wheel is fixed to the stem by a threaded hand wheel nut. The arrow pointing the direction to open the valve will be marked with the word "open" or "close" or "shut", unless the size makes it impracticable. Valves shall be closed by turning the hand wheel in clockwise direction.

The material of construction of hand wheel shall be malleable iron, Carbon steel, Nodular iron or Ductile iron. Cast iron is not preferred. The nut shall be of carbon steel or stainless steel.

When the installed position of the valve is such that the hand wheel is not accessible, then the hand wheels are replaced by chain wheels and the valve is operated with chains. For large diameter valve where the operating torque is high, gear arrangement is provided to facilitate operation. Mostly, bevel gear equipment is adopted. General recommendation for specifying Gear operator is:

Valve Rating	Size
Class 150	14" NB & above
Class 300	12" NB & above
Class 600 & above	8" NB & above

If remote operation of the valve is required, then this could be achieved through motor with limit switches. Proper selection of the drive unit should be done depending on the services.

1.1.9 Bolting

Normally high tensile stud bolts are used for bonnet bolts and low carbon bolts for gland and yoke bolting. Gland bolts are normally hinged bolts with hexagonal nuts.

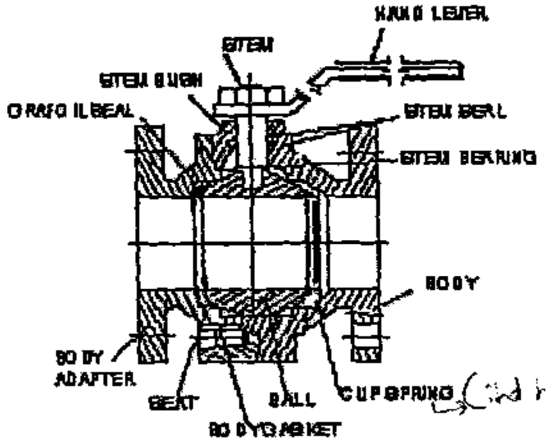
1.1.10 Valve Port

The valve size is specified by the size of the end connection or the body end. The port or the bore is the passage through the valve.

There are two types of port designs possible in gate valves, full bore and reduced bore. In case of full bore, the net area of the bore through the seat shall be as nearly practicable equal to the nominal pipe size. For reduced port valves, the port diameter is normally one size less than the size of the end.

The compact design small bore (½ - 1 ½ inch) gate valves are as per API 602 or BS 5352. Unless the full bore design is specifically asked for, manufacturers supply the reduced bore valves. The full bore design gate valves are also covered in BS 5352 and is designated as 'std bore'. In full bore design, the net area of the bore through seat shall be equivalent to that of Sch 80 pipes for class 800 valves and Sch 160 pipe for class 1500 valves. In no case less than 90% of the above figure is acceptable as per code.

1.2 BALL VALVES



The ball valves are normally used as positive shut off valves. The positive shut off is attained because of the soft seats. Special design is also available with ball having shaped port for regulation. Metal seated ball valves are also available for high temperature service. The ball valves can be classified based on:

In case of ball valves - If ball is with cavity, pressure relief facility shd be provided.

- The port size
- The type of body construction
- The construction of seat
- The construction of ball

The construction of stem

The above classification is in addition to the ones based on the end connections, material for construction and the pressure classes. **The pressure temperature ratings** of the ball valves are generally established by the materials of the seat rings. The service temperatures are also limited by the material of seat rings.

The ball valve offers minimum resistance to the flow. There are two types of designs available as far as the flow area through the valve is concerned. They are the Full Port design and the Regular Port (Reduced Port) design. In full port valves, the port diameter will be equivalent to the nominal size of the valve, whereas in the regular port valves, the port diameter will be one size smaller than the nominal size. Valves with shaped port are used for flow control applications.

Based on the body construction the valve could be classified as:

- Single piece design
- Two piece design
- Three piece design
- The short pattern
- The long pattern
- Sandwich design.
- Flush bottom design.
- Extended body design.

Lined Body

In the single piece design valve, the body will be cast/forged as one piece. The insertion of the ball will be through the end or through top of the body and is held in position by body insert or bonnet. The side entry design restricts the valve to be of regular port only.

In two-piece design, the body is constructed in two pieces and the ball is held in position by body stud. There can be full port or regular port design possible in this construction. In case of three-piece construction, the body has two end pieces and one centrepiece. These are held by body studs.

The three-piece construction permits in-line servicing without disturbing the existing pipe work. If the valves have socket weld, screwed or butt-welding ends, this design totally dispense with the necessity of companion flanges.

The short pattern and the long pattern of the body is on the basis of the end to end dimensions. Normally short pattern body is adopted by the manufacturer up to 300 NB valves for 150 LB class. In case of 300 NB to 400 NB, class 150 short pattern valves, the ball may protrude beyond the body and flange faces when the valve is in closed position.

The sandwich design is the flangeless design adopted by some manufacturers. This is to confine the use of the high cost exotic materials like Alloy-20, Hastelloy-B, Hastelloy-C, etc. to the wetted areas only. The valve is designed to fit between the flanges. The body cover gets bolted to body with studs or hexagonal head screws.

The seat rings are renewable in the ball valves except for those having one-piece sealed body construction. The two different types of seat construction

are possible, viz., the fire safe design and the non-fire safe design. In the fire safe design, a secondary metal seat will be provided so that when the soft seat is fully burnt, the ball will shift its position and seat against secondary metal seat and arrest full leakage. The modified design incorporates a double staged stem seat design and a seating system that adjusts to the line differential pressure. At low differential pressures the floating ball seats against resilient tip seat. At higher differential pressures, the ball deflects to produce contact across the entire seating surface of the seat ring.

In an actual fire, the heat intensity of the fire could be so different that it is impossible to ensure that elastomer seats are fully damaged during fire. If the seats are only partially damaged, the ball cannot take seating against the secondary metal seat and hence the valve would leak. Hence, in my opinion, none of the soft-seated ball valves can be declared fire safe since the valves are bound to leak in case of partially damaged seats. The manufacturers have come up with metal-seated ball valves, which are fully fire safe. Here the resilient seats are replaced by metal seats, which could even be deposited with high temperature resistant materials. The fire safe design should also ensure that any development of static electricity should be fully discharged by proper design and manufacture of valve. Such an arrangement is called the 'Anti-static' design. This ensures to have a discharge path from ball to the spindle and from spindle to the valve body with an electrical resistance of not greater than 10 ohms when the valve is new. A typical method of achieving earthed continuity is to provide stainless steel spring-loaded plungers, one fitted between the stem tongue and ball and second fitted between stem and body.

The ball could be of full bore or a reduced bore. The design aspect of the same has been explained earlier. The ball at the bottom end of the body could be supported fully by the seat or it could be *trunnion supported*. The ball can be solid ball or of hollow construction with cavity. The cavity is to be sealed when the valves are used in volatile liquid. This design of the ball is called sealed cavity design.

The gland shall be bolted type or screwed. Internally screwed stuffing box is not allowed by code. Bellow sealed bonnet is also provided in case of valves used in lethal services. Two basic bellow seal designs are available. The same is explained under plug valves.

The valves shall be operated by wrench or by hand wheel with gear arrangement. The wrench shall be designed so that it is parallel to the flow passage of the ball. The valve shall be closed by turning the wrench or the hand wheel in clockwise direction. The length of the wrench or the diameter of the hand wheel shall be such that minimum force is required to operate the valve under the maximum differential pressure.

When added emission control is required, additional packing and leak off port are options that can be added.

Normally all the parts are metal except the resilient seats in a ball valve. Plastic valves are also selected for corrosive process fluids while they operate up to 150 psi and 100-150°C and also in food industry. To select the best plastic valve, process data such as number of cycles before failures is critical. Ball valves lined with PTFE on the body and ceramic ball is used for extreme corrosive fluids.

1.3 PLUG VALVES

The plug valves, like ball valves, are quarter turn positive shut off valves. Two major types of plug valves are in use. They are the lubricated metal-seated plug valves and Teflon sleeved plug valves. These valves can have flanged, butt-welded, screwed or socket

weld ends. The pressure classification is the same as that specified for the gate valves. The range of pressure to which these valves could be used depends upon the seat, seals and the lubricant. Plug valves with shaped port are used for flow control applications.

1.3.1 Metal Seated Plug Valves

In lubricated plug valves, the lubrication of the seating surface is by means of lubricant, which is fed into the operating surface of the valve either in the form of mastic sticks or by grease gun. The selection of the lubricant depends upon the service to which the valve is subjected to. In certain designs, a low friction Poly Tetra Fluoro Ethylene (PTFE) is impregnated on the surface structure of the valve plug. This is called 'LOMU' treatment. This reduces the frequency of valve lubrication.

The plug valve design refers to three patterns considering the shape or port through the valve and the overall length. They are the regular pattern, the short pattern and the venturi pattern.

The regular pattern valves have plug ports generally rectangular in section and have area substantially equal to full bore of the pipe. The transition from the round body to rectangular seat ports is smooth without sudden alteration in section, which causes turbulence. These are used where pipeline losses are to be kept minimum.

The short pattern valves have face-to-face dimensions corresponding to wedge gate valves. This is used as an alternative to gate valves.

The Venturi Pattern Valves have reduced port area. The change of section through the body throat is so graded as to produce a venturi effect to restore a large percentage of velocity head loss through the valve and produce a resultant total pressure drop of relatively low order.

The plug could be installed with the taper towards the bottom end of the body or reverse. When the installation is with the tapered portion towards the top, it is called 'inverted plug'. Normally larger diameter (8"NB and above) have this design.

Another design in use is the Pressure Balanced Plug. The benefit of the pressure-balanced design is the elimination of the possibility of unbalanced forces causing taper locking of the plug. This is achieved by using the live line pressure to replace the sealant pressure. The regular sealant injection is not needed to keep the valve free to turn.

The pressure balance system consists of two holes in the plug connecting the chambers at each end of the plug with the port, which contains line pressure.

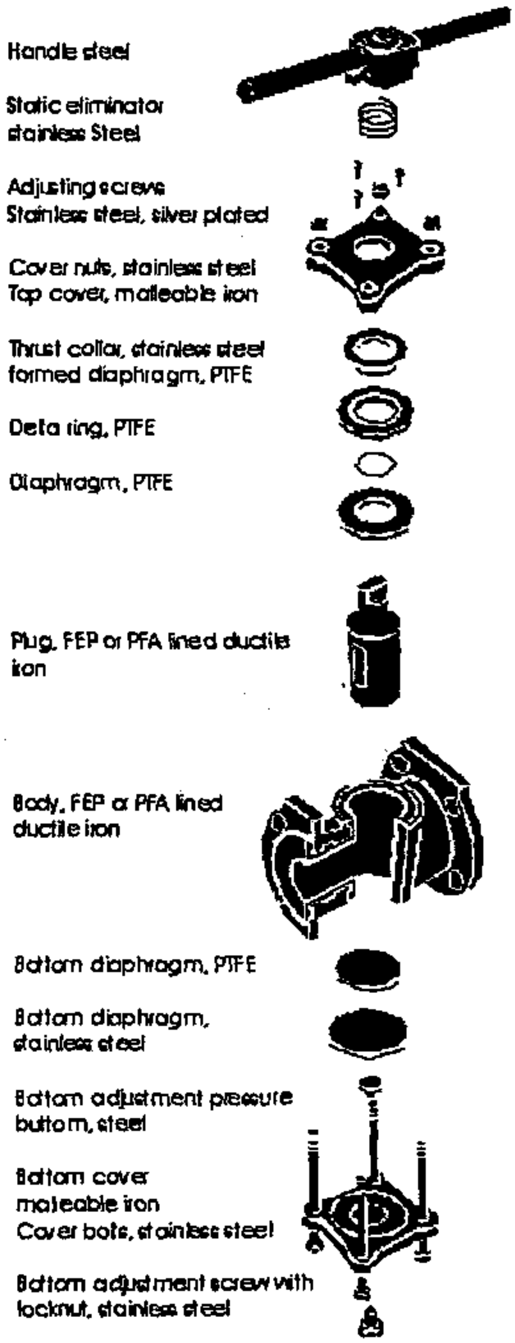
The valve having pressure balanced is called dynamically balanced plug. M/s Audco called these types of valves as 'Super-H' pressure balanced valves. The break away torque required to operate these valves are lower than (almost half) that for the reduced port ball valves.

Comparison of Breakaway Torque Requirement of Valves

The following data has been published by a valve manufacturer to indicate the easiness in operation of the pressure balanced plug valve.

Size (NB)	Full Bore Ball	Reg. Bore Ball	Pr. Balanced Plug
2"	73	86	60
3"	284	284	120
4"	1158	284	175
6"	1158	1158	325
8"	1950	1158	600
10"	--	--	910
12"	--	--	1300

1.3.2 Teflon Sleeved Plug Valves



In Teflon sleeved plug valves, the plug and the body in the valve are separated by a PTFE Sleeve. This sleeve serves as the seat for the valve plug, thus eliminating the contact of two metal surfaces. Here, the turning effort is low and friction is avoided. The limitation is the temperature to which the sleeve can be subjected to. The sleeved plug valve also is available complying with the fire safe atmospheric seal. They are not manufactured fire safe through seat. The anti static design as explained for ball valve is also possible in Sleeved plug valves.

The sleeved plug valves are also designed with bellow seals to control the emission rates. There are two basic bellow seal design for quarter turn valves. One is the "goose neck" or the "bent-straw" design. The other is the "rack and pinion" type. The rack and pinion type maintains a linear bellow so there is less stress and no forging. There is an alternative to bellow design is available and is called a 'caged' plug valve. In this design the plug is inserted in another plug and it provides inherent emission control characteristics of the sleeved plug valve while improving the throttling capabilities and reducing wear potential. Plug valves are also available with Fluoro polymer lined metal body and plug.

1.3.3 "Permaseal" Plug Valves

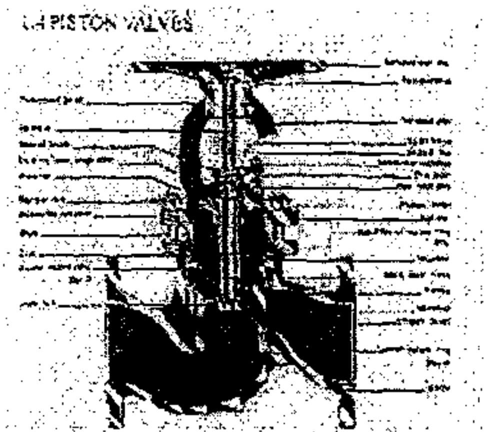
These valves are similar to the sleeved plug valves but are provided with Teflon seats instead of sleeves as in the case of ball valves. These are designed for on-off applications and can handle clean viscous and corrosive liquids. The construction features and operation are identical to that of the sleeved plug valves. Graphite seats also can be provided for high temperature service. But this design cannot provide drip-tight shut off.

1.3.4 Eccentric Plug Valves

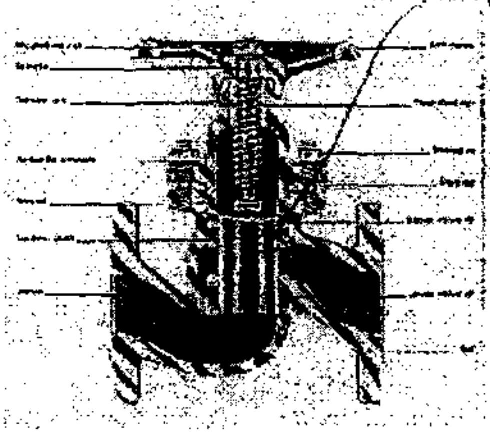
These valves are provided with plugs, which are mounted off-centre. Eccentric

plug valves are used in corrosive and abrasive service for on-off action. Eccentric action plug moves into and away from seat eliminating abrasive wear. These are covered under MSS-SP standards.

1.4 PISTON VALVES



Piston



Piston valves resemble in construction more towards a globe valve and are used for shut off and regulation. These valves provide positive shut off. The shut off assembly comprises of a metal piston, two resilient valve rings and a metal lantern bush. The sealing surface consists of the outer vertical surface of the piston and the corresponding inner surfaces of the sealing rings. This provides a large sealing surface compared to globe valves of conventional design.

Piston valves are of two types, balanced and unbalanced. Balanced valves are used in high-pressure services and unbalanced one for low-pressure services.

The main parts of the valve can be identified as

1.4.1 Body

1.4.2 Bonnet

1.4.3 Piston

1.4.4 Valve rings

1.4.5 Lantern bush

1.4.6 Spindle

1.4.7 Gland

1.4.8 Packing

1.4.9 Hand wheel

1.4.10 Yoke bush

1.4.11 Bonnet stud

1.4.12 Gland eyebolt

The body is normally of cast construction. It can have screwed ends, flanged ends or butt-welding ends. These valves follow the regulatory codes to DIN. There are no API or ANSI standards covering the piston valves. The end-to-end dimensions are to DIN 3202, which is more than a gate or globe valve of the same size to API/ANSI/BS standards. Of late, the Piston Valves are also made to ANSI B16.10 dimensions. The end connections are also available to ANSI/BS standards.

The bonnet is also of the same material as that of the body and it is of bolted construction. The piston along with the two resilient seats provides proper sealing. The upper valve ring seal to atmosphere, the lower valve ring provides seal across the ports. The lantern ring serves as the distance piece between the two rings.

There are two types of piston designs available: Regulating type and the normal. In regulating type the bottom part of the piston is tapered to have throttling effect. The sealing rings are the heart of the piston valves. The sealing rings are made from specially developed high quality elastomer material or graphite. The materials are selected depending upon the service conditions viz. The fluid for which the valve is used and its pressure temperature conditions.

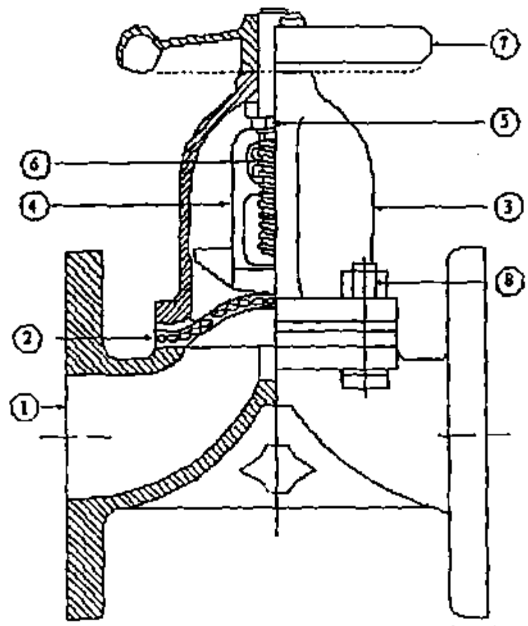
Spring washers are fitted under the bonnet nuts to ensure that the pressure of the bonnet on the valve ring is kept constant. This along with the resilient sealing rings produces a spring action, which compensates for any differential expansion that can occur.

There are two types of stem designs available, the inside screw rising stem and the O, S and Y type with rising stem. The hand wheel is of rising design. In O, S and Y type, a stuffing box with a bolted gland is provided. This design is mainly used for Thermic fluid/High temperature services.

The piston valves are preferred by maintenance people, as they need lesser attention. They call it as 'Fit and Forget' type of valve.

1.5 DIAPHRAGM VALVES

Diaphragm valves are mainly used for low-pressure corrosive services as shut-off valves. These can also be used as control valves. Here the diaphragm moves up and down to operate the valve. The valve body can be lined or unlined.



Lining material is selected to suit the corrosive nature of the service fluid.

Diaphragm valve with plastic body is also manufactured.

There are two types of diaphragm valves available. They are the 'Weir' type and 'Straight flow' type. The most commonly used one is the weir type and are popularly known as the 'Saunders' type. In this type, the body configuration is such that isolation as well as control is possible.

A typical diaphragm valve has the following major parts that could be identified. They are:

1. Body
2. Diaphragm
3. Bonnet
4. Stem
5. Stem bushing
6. Compressor

7. Hand wheel

8. Bonnet bolting.

The body and the bonnet are made of casting. The material of construction of the body depends upon the service for which it is used. The body can also be lined with corrosion resistant materials like PTFE, Glass, Rubber, etc. depending upon the corrosive nature of the fluid or could be entirely made out of plastic material. The diaphragm is normally made from an elastic material like PTFE or rubber. The diaphragm presses against the body to give positive shut off. The port can also be adjusted by controlling the position of diaphragm, which is being done for control applications. The diaphragm is secured between the bonnet and the body. The compressor attached to the diaphragm facilitates the up and down movements. There are two types of stem designs possible in a diaphragm valve. They are the 'Indicating' and 'Non-indicating' type. In the indicating type, the position of the spindle indicates the port opening. The opening and closing of the valve is effected by the hand wheel in a manually operated valve. The material of construction of the hand wheel could be ductile /malleable iron or even plastic. The body ends could be flanged, screwed or butt-welded as required. In case of diaphragm valve with lined body, the ends are always flanged and the lining extends to the flanged surface.

The use of these valves is restricted as they can withstand a maximum operating pressure of 7 to 10 kg/sq.cm g. The damage to the diaphragm occurs and hence the maintenance is more frequent. On lined valves, spark test is also conducted in addition to the pressure tests. This is to ensure that the lining is continuous and no 'holiday' occurs.

There are no API or ANSI standards for this type of valves. These are covered by British Standard and MSS-SP Standards.

1.6 BUTTERFLY VALVES

Butterfly valves are positive shut off quarter turn valves. The major parts of the butterfly valves are:

1. Body

2. Disc

3. Shaft

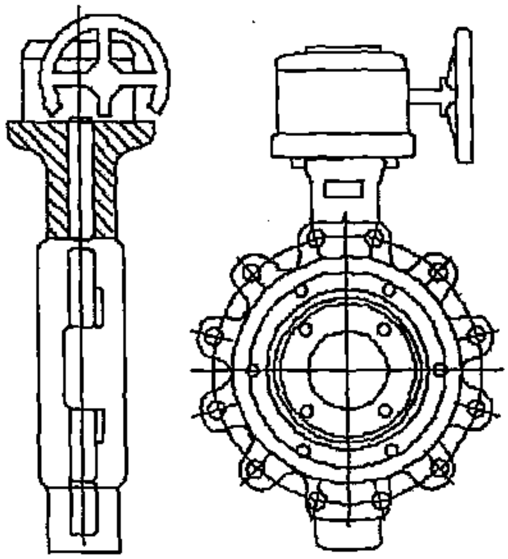
4. Body seat

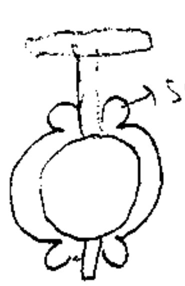
5. Disc seat or seal

6. Shaft seal

7. Shaft bearing

8. Handle.





Space for bolts for flanges on both sides

There are three types of body designs possible in a butterfly valve. They are the double flanged type, wafer lug type and wafer type. In the double-flanged body design, the disc is contained within the body and is fitted to the pipeline like any other conventional valve. These types of valves are used rarely as the advantage of sandwich design is not available with the same.

In the wafer lug type and wafer type, the valves are designed to permit installation between ANSI/BS/DIN flanges. There are different designs available in these types. In certain designs, the body is lined with a resilient material such as Nitrile rubber, Ethylene Propylene Diene Monomer (EPDM), PTFE. The metallic disc with or without coating ensures proper sealing against these liners. By selecting proper disc material, this type of valve can be used for corrosive services. The body could be of any material. There is no gasket needed for the installation of these valves.

In certain other designs, the body will be provided with soft seat instead of a liner. This seat flexes against the sealing edge of the disc when the valve is closed. The seat is made of PTFE with certain reinforcements. This seating is designed to replace the PTFE seats when worn out.

Another design is the offset shaft and eccentric disc design, which imparts camming action to the disc. In this, the stem centre line, the disc centre line and the pipe centre lines are offset. This feature causes the disc to swing completely out of contact with the seat upon opening, eliminating wear points at top and bottom of seat. On closing, the disc moves tightly into the flexible lip for reliable seating around the entire seat.

The difference between the wafer lug and wafer type body design is that the former has provision for all the studs to pass through the body whereas the latter has provision for only locating bolts. The wafer lug design is also called single flange design.

As regards the shaft is concerned, there can be a single shaft or a main shaft and a stub shaft at the bottom of the disc. Single shaft is a better design as it minimizes the deflection. The shaft sealing can be done with 'O' ring or stuffing box and packing. These valve designs provide inherent emission control advantages over rising stem valves.

Valves up to 12" NB are operated with lever. The lever can have positions to control the flow. Higher diameter valves are provided with gear unit and hand wheel. When used as control valves, these can be provided with actuators also.

The use of this type of valve for high temperatures is limited by the material used for seats. Only resilient seats can provide positive shut off. Metallic seating can also be provided for use at higher temperatures but will not provide positive shut off.

These valves can be used for vacuum service. When used for cryogenic service, the valve shall be provided with extended shaft to clear the insulation.

When used beneath a hopper for solid handling applications, tight shut off is troublesome since particles jam between valve closure surfaces. Further, the valve must be strong to lift half the disc against the weight of the solids in the hopper. The advantages of these valves are that the wear resistant elastomer has a longer life expectancy than the conventional metallic seated

(even stellited) valves when used in high-density mineral slurries. The seating problem in other type of valves does not affect these valves as the encrusted scale will break when the valve operates and solids flush away with the flow. The sleeve is the only wetted part and by selecting the right sleeve material, the valve body can be made out of low cost material. As the design calls for no gland, there is no fugitive emission and meets the EPA requirements.

1.7 PINCHVALVES

Pinch valves are also similar to diaphragm valves. In Pinch valves, the bodies provided with sleeves, which get squeezed to control or stop the flow. The sleeve could be of corrosion resistant materials like Rubber or PTFE. The body is normally made from cast iron. These are used for special services where service pressures are very low like isolation of the hose connections etc. in the chemical process industry. The body is cast and can have flanged or screwed ends.

Of late manufacturers have developed these valves to endure higher pressures and temperatures (0 to 100 bar & up to 120°C respectively) for application in mining and mineral industry.

These valves are also not covered under API or ANSI standards and are manufactured as per Manufacturers' standards.

2.0 REGULATING VALVES

The valves normally used in the plant to regulate/control the flow are:

2.1 Globe Valves

2.2 Needle Valves

2.3 Butterfly Valves

2.4 Diaphragm Valves

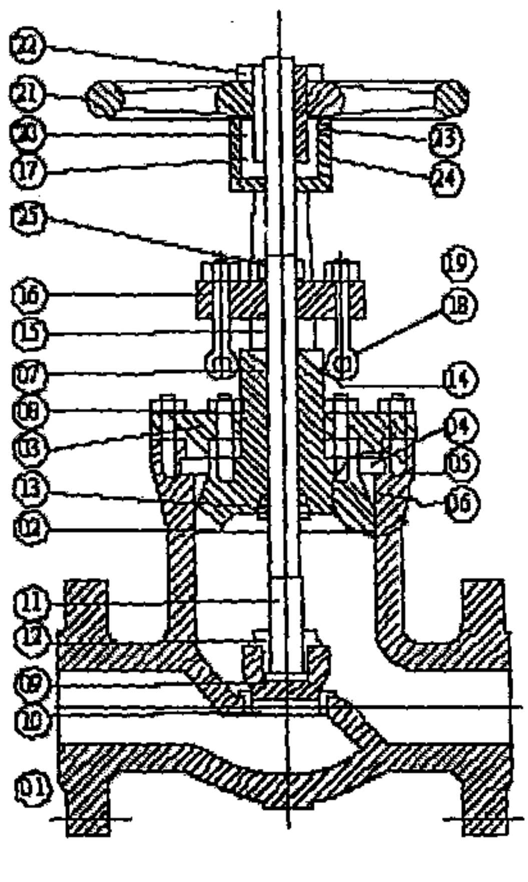
2.5 Piston Valves

2.6 Pinch Valves

The features of Butterfly, Diaphragm, Piston and Pinch valves were already explained under isolating valves. There are many identical features in the construction of gate and globe valves. The foregoing note is intended to explain the comparison between these valves highlighting the differences.

2.1 GLOBE VALVES

A typical globe valve has the following parts, which could be identified.



1. Body
2. Bonnet
3. Yoke
4. Backup Ring
5. Thrust Ring
6. Gasket
7. Gland
8. Stud and Nut
9. Plug
10. Seat Ring
11. Spindle
12. Plug Nut
13. Back seat
14. Clamp
15. Gland Bush
16. Gland Flange
17. Yoke Sleeve
18. Cross Bolt and Nut
19. Eye Bolt and Nut
20. Yoke Nut
21. Hand Wheel
22. Hand Wheel Nut
23. Grub Screw
24. Grease Nipple
25. Anti-Rotation Device

2.1.1 Body

The construction of the body differs from that of the gate valve. The body ports are arranged such that the flow is from the underside of the disk. Though the code specifies that the globe valves shall be designed suitable for installation in either direction of flow, the preferred direction of flow for globe valve shall be from under the disk. Normally the direction of flow is cast or embossed on the valve body.

There are two types of port designs possible, the full port and the reduced port. In the full port design the body ports shall be as large as practicable design considerations permit. However, in no case the net area of the bore through the seat of globe shall be less than the 85% of the area of the actual pipe bore. In the reduced port design, the port diameter is normally one size less than that of the connected pipe.

2.1.2 Bonnet

The body bonnet connection for the globe valve is the same as that of gate valves.

2.1.3 Disk

The disk of the globe valves shall be:

- Flat faced type
- Plug type
- Ball type
- Needle type
- V port type

The flat-faced type disks are used when the valve is to be used for the positive shut off service. For such

based on the pressure drop across the valve we have to decide the size of the valve (CV)

valves, disk can be provided with an elastomer ring or facing which will ensure the same. The needle type disks are used when finer flow control is to be achieved. These disks can be also of contoured design as used in flow control valves. These are generally used for precise flow control applications. V port type disc is used for throttling application.

The disk shall be either loose or integral with stem. The integral design is used mainly for the needle type of disc. The loose plug design allows the same to be renewable. When in the fully open position, the net area between the disk and the seat shall be equal to the area through the seat.

Bellow seal is the only way to achieve emission control in this type of valve.

2.1.4 Stem

In case of globe valves, the stem is always of rising design along with the hand wheel. The stem is provided with a disk nut at the lower end. The upper end is provided with a hand wheel screwed by stem nut. In case of bellow sealed valves rising stem with non-rising hand wheel is provided similar to that in the case of gate valve.

2.1.5 Gland, Gland flange, Packing & Lantern

Design and details same as that of gate valves.

2.1.6 Seat rings

In case of globe valves of carbon steel, the hard faced seats can be directly deposited on the body or the seat rings shall be shoulder seated.

2.1.7 Yoke and Yoke bush

The construction of the Yoke is the same as that of the gate valve. The Yoke sleeve of the gate valve is machine finished on all surfaces whereas that of the globe valve shall be screwed or fitted in the position and locked in case of rising stem design.

2.1.8 Hand Wheel & Hand Wheel Nut

Unlike in gate valve the hand wheel also rises along with the stem for globe valve. When used as a control valve, actuators are fixed so that the stem movement is effected through the same. In case of bellow-sealed globe valves, the non-rising hand wheel design is provided similar to that of gate valve. This is to ensure that the bellows are not subjected to torsion.

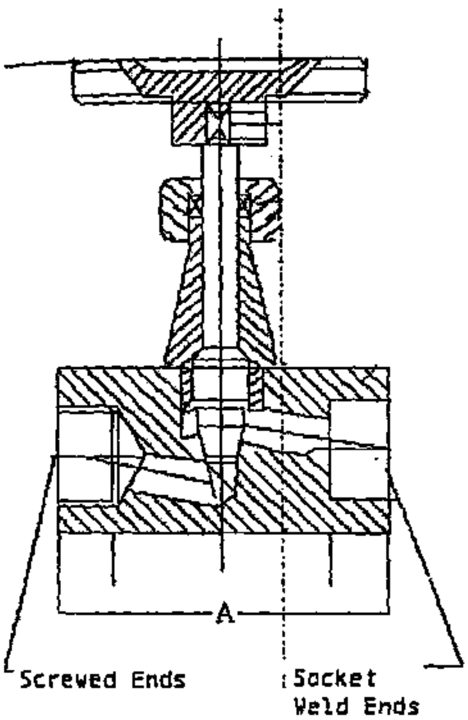
The above are the major design aspects of the globe valves and comparison of the same with that of the gate valves. As regards the material of construction, end connection etc. are concerned, the same shall be selected by the piping engineer based on the service of the line to which the valves are used. The environment in which the valve is installed also will have to be considered while selecting the material

There could be slight variation in design from manufacturer to manufacturer, but the basic design features as specified are not altered.

2.2 NEEDLE VALVES

The needle valves, like globe valves, are used for flow control. Normally needle valves are used in smaller sizes and are provided with either screwed or socket weld end The design of the needle valve can be exactly same as that of the globe valve except for the disk. In globe valves, the disk is

like a truncated pyramid whereas in the needle valves it will be full. This facility eures finer flow control. The disk could also be integral with the stem, in which case the bottom part of the stem will be machined accordingly.



A totally different type of construction is also used for the needle valves of smaller sizes. The body/ bonnet connection will be screwed on type instead of bolted. In place of a flanged gland with gland bolting, the packing will be positioned with a screwed union gland nut. The stem will be of inside screw arrangement. This makes the valve compact.

The body and bonnet can be of forged construction or can be fabricated from barstock.

These valves are used only for limited applications. Even though the code covers this design, these are mostly made as per Manufacturers' standard.

2.3 BUTTERFLY, DIAPHRAGM, PISTON AND PINCH VALVES

The design and the construction features of the same are already explained under the head 'Isolation Valves'. These valves can perform the dual duty of control as well as isolation.

3.0 NON - RETURN VALVES

As the name indicates, these valves are used to ensure unidirectional flow of fluids. Check valves are mainly divided into two types based on check mechanism.

3.1 Lift check valves

3.2 Swing check valves

The type is selected depending upon the service, size and material of construction. Normally, small bore valves (up to 40 mm NB) are selected as lift check and big bore as swing check due to constructional limitation.

3.1 LIFT CHECK VALVES

These valves operate by the lifting action of the disk/element. The different types of lift check valves available are –

3.1.1 Piston lift check

3.1.2 Ball lift check

3.1.3 Non-slam check (vertical position)

3.1.1 Piston lift check

The piston lift valve has body similar to that of globe valve. The piston

D check valves are used in horizontal positions,
In vertical position, they can be used with spring

will be in cylindrical form, the lower end of which is shaped to form a seating disk. The cylindrical part fit into the guide making an effective dashpot. When it is in fully open position, the net area between the seating disk and the seat will be equal to the area through the seats.

The body will be provided with renewable body seat rings like in globe valves. In carbon steel valves, there can be hard faced seats deposited directly on to the body.

The piston lift check valves can only be placed in the horizontal pipeline.

The lift check valves can also be provided with spring-loaded piston. In this case, a spring of specified tension has to be placed, between the guide and the piston within the cylindrical portion. This type can be placed in any location.

3.1.2 Ball lift check

In ball lift check valves the unidirectional flow is achieved by the movement of a ball. There are two designs possible in this pattern, the horizontal and the vertical. In vertical design, the valve should be placed in such a way that the flow is always in the upward direction.

These check valves are provided with guides to guide the ball throughout the travel. The travel should be such that in fully open position, the net area between the ball and the seat shall be at least equal to the area through the seat.

The main parts of lift check valves are the following.

1. Body
2. Ball / Piston
3. Cover
4. Seat
5. Guide
6. Gasket
7. Cover stud nut

The body shall be of forged or cast construction and with socket welded/screwed/flanged ends, integrally cast or with welded-on flanges.

The cover shall be either bolted or welded or with union nuts. The union nuts could be of hexagonal or octagonal shape. The cover material shall be same as that of the body.

The seating shall be integral or renewable. The hardness difference can also be achieved by weld deposit on seating surfaces. The renewable seat rings shall be screwed-in type either shoulder seated or bottom seated.

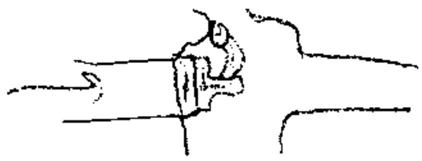
3.1.3 NON-SLAM CHECK VALVES

disc

The non-slam check valve is a spring loaded lift check valve with a modified design of the body. The valve is designed in such a way that the same can be sandwiched between the two flanges. Here the disc is held in position by a spring, which is housed, in a housing cap or yoke. can be used in vertical applications

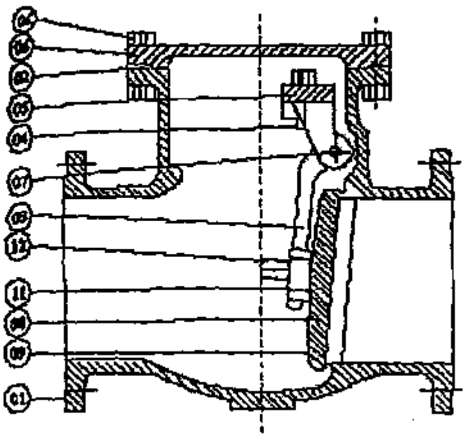
3.2 SWING CHECK VALVES

These valves operate by the swinging action of the disk. There are two types of swing check valves available. They are the conventional swing check valves with flanged ends and the wafer type spring loaded check valves.



3.2.1 Conventional Swing check valves

In these types of valves, the check mechanism is the disk, which is hinged. The pressure of the fluid lifts the disk and allows the flow. The disk



returns to the seat with its own weight. This allows the valve for mounting in horizontal as well as vertical position with upward fluid flow. The main parts of the valves are -

1. Body
2. Cover
3. Hinge
4. Hinge Bracket
5. Gasket
6. Cover Stud and Nut
7. Bracket Stud and Nut
8. Disc
9. Seat Ring
10. Hinge Pin
11. Disc Pin
12. Washer

The body will be cast with a tapered wedge seat and will be provided with renewable seat rings. The wall thickness and end to end/face to face dimensions of the body shall be as per the regulatory code to which it is designed. The end flanges shall be integrally cast or attached by welding. The flanged connection shall be to ANSI B 16.5 or any other flange standard. The butt-welding end connection shall be to ANSI B 16.25.

The disk will be attached to the body through hinge and hinge pin and swings against the same controlling the flow. The disk material shall be of quality at least equal to that of the body.

The cover will be bolted on to the body. The bolted connection shall be raised face/ tongue and groove/male and female/ring type joint depending on the pressure rating of the valve. The gasket shall be selected to suit the type of connection. It can be corrugated or flat solid metal, corrugated or flat metal jacketed, asbestos filled, metal ring joint, spiral wound asbestos filled. Flat ring compressed asbestos is used for low-pressure application, Teflon or Teflon filled for corrosive applications. Normally high tensile bolts are used for cover bolting. In cast iron check valves low carbon steel bolts are used.

3.2.2 Wafer check valves

The wafer check valves are the flangeless swing check valves. These are covered under the regulatory code API 594. There are two types of wafer check valve designs available.

a) Single plate wafer check valve

b) Dual plate wafer check valve

The arrangement of single plate check valve is somewhat similar to the conventional swing check valve. Here a circular plate seated against the valve body seat by line backpressure or flow reversal acts as a valve closure. This is further aided by the provision of spring.

In dual plate check valves, there are two spring loaded semi circular plates. The plates are arranged in such a way that the spring force acts beyond the centre of area of each plate and the fluid force acts within the same. This fulcrum causes the heel to open first preventing rubbing of the seat surface prior to normal opening. The sizes specified in API 594 are from 2" NB to 48" NB. Manufacturers have developed standards beyond these sizes as well.

The plates shall be made of material at least equal to that of the body. The body and plate seating surface can be renewable or integral or with deposited metal. The seat surface could be stellited or can be of resilient material. In these valves, the items specified under trim are the seating surfaces, springs, hinge and bearings. Table 4 of API - 594 gives trim numbers and the corresponding material of construction.

Compared to the conventional check valves, these have less pressure drop across the valve in larger sizes, reduced water hammer and are compact.

4.0 SPECIAL-PURPOSE VALVES

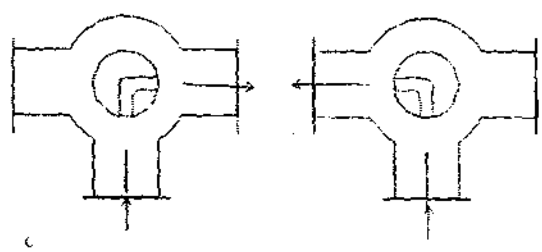
Valves, which perform duties other than the two-way isolation, control and check, are classified under the category of special purpose valves. Few of such valves are

4.1	Multi-port Valves
4.2	Flush Bottom Valves
4.3	Float Valves
4.4	Foot Valves
4.5	Line Blind Valves
4.6	Knife gate Valves

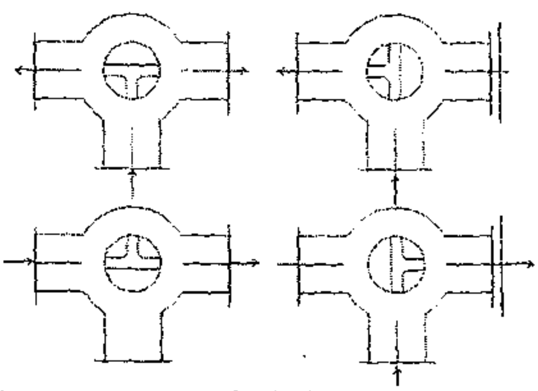
4.1 MULTI-PORT VALVES

Any valve, which has more than two ports, is classified as Multi-port Valves. The multi-port valves on certain services reduce the time for operating and the over all costs. There are three port valves and four port valves in common use. Five-port designs are also available. Two types of three port designs are available viz. The 'T' port and the 'L' port. The possible flow patterns of these are as below:

L PORT VALVE



T PORT VALVE



A most economical layout could be selected from the study of above flow patterns.

The typical applications of the three way valves are-

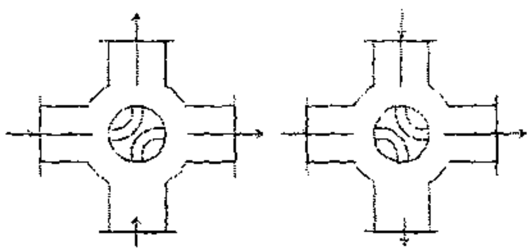
(1) alternate connection of the two supply lines to a common delivery,

(2) diversion of flow to either of two directions,

(3) isolation of one of a pair of safety valves for maintenance purpose,

(4) division of flow with isolation facility.

The flow patterns of a four-way valve are –



The typical applications of four way valves are:

(1) Reversal of pump suction and delivery

(2) By pass of strainer or meter

(3) Reversal of flow through filter, heat exchanger or dryer.

The types of valves used for this design are the ball or the plug valves. However, globe pattern valves also can be designed with suitable disc positions to achieve the three-port design.

The advantages of multi-port design valves are -

(1) Reduction in number of valves used,

(2) Quick and easy operation,

(3) Simplification of piping layout and thus economy in fittings,

(4) Less risk of product mixing by incorrect valve operation,

(5) The stops can be arranged to arrest the unrequired flow patterns and at the same time make it impossible for desired positions to be obtained.

Two of the multi-port valves can also be inter coupled to permit fast multiple operation in the simplest possible way and with minimum manpower.

It is essential for the designer to specify the exact requirement of flow patterns based on the piping arrangement to the manufacturer. Lack of proper coordination will result in a totally different output than what is required.

4.2 FLUSH BOTTOM VALVES

These are special type of valves, which are used to drain out the piping, reactors and vessels. These are attached to the vessels on pad type nozzle. The disks in closed position match with the bottom of the vessel or piping leaving no room for hold up or stagnation.

There are two types of flush bottom valves.

a) Valves with disk opening into the tanks.

b) Valves with disk into the valve.

In the first case, the stem pushes the disk into the tank to drain the liquid. This type cannot be used when there are any internals, which restrict the movement of the disk. The draining of the material could be effected

completely. In the second case, the disk gets pulled down into the valve effecting the discharge of material.

There are two types of disk design available, the plug type and the ram type.

Normally, the inlet size of a standard flush bottom valve is one size higher than that of the outlet size. There are special constructions possible with both sizes same. The outlet port is at an angle to the inlet port. Normally 45 or 60 degrees deviation is provided. The end connections are normally flanged. However, smaller size sample/drain valves have been developed with welding end at inlet to withstand higher pressures. The maximum rating available at present for flanged valves is ANSI 300 lbs.

The parts of the flush bottom valve are identical to that of a globe valve and the closing and opening actions are also similar. The shut off is achieved by disk closing against the body seat. The disk could be Globe type or Ram type.

Jacketed flush bottom valves are also possible if required for the service. The disk and seat also could be machined to such accuracy to serve the vacuum duty as well.

4.3 FLOAT VALVES

Float valves are used to control the level of fluid in a reservoir. Only the inlet of the valve is connected with the supply pipeline and the outlet is open to the reservoir. There will be a float with lever, which controls the movement of the piston regulating the flow.

These valves are covered under the Indian Standard IS 1703. There are two types, the "HP" and the 'LP' depending upon the pressure holding capacity of the valve.

The body lever and internals are manufactured out of gunmetal and the float is of PVC or copper depending upon the temperature of the fluid. The lever length could be adjusted to suit the level in the reservoir.

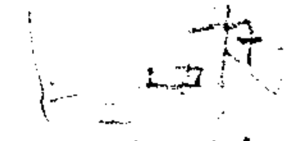
These valves have threaded ends and are connected to the wall of the reservoir with hexagonal nut. The reservoir need not be provided with a nozzle, only an opening is required. The level of liquid will always be below inlet connection.

The maximum size of valve covered under the standard is 50 NB and special design has to be done if higher size valves are required. These are called equilibrium float valves.

4.4 FOOT VALVES

Foot valves are a sort of non-return valves with strainers mounted at the open end of the pump suction pipelines. These are used when the pump has negative suction. The check action of the valve holds the priming fluid of the pump while the pumps are filled before starting. The suction strainer helps to hold the solids while the pump is sucking the fluid.

These valves are covered under the Indian Standard IS 4038. There are two types of check mechanisms available viz. the lift check and the swing check. The operation of this is similar to the Non-return valves. Valves are available with either flanged end connection or screwed end connection. The material of construction of the body is Cast Iron or Gunmetal for the valves.

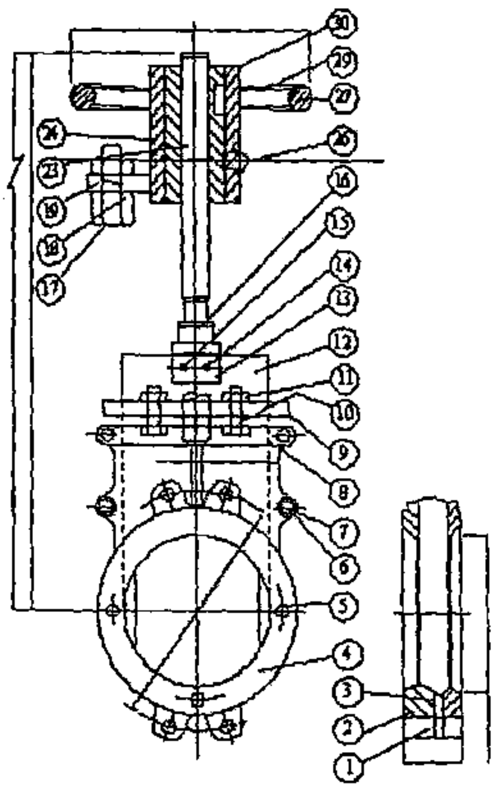


level ↓ liquid in

level ↑ liquid

4.5 LINE BLIND VALVES

Line blind valves are used for positive shut off and replaces a 'spectacle blind'. Unlike blind plates these are easier to operate and less expensive than a standard valve. The most common type of the line blind valve is three-bolt/five bolt line blind valve. Another design is the single gate line blind valve or the 'pulp valve' which can be power operated or with hand wheel. There also exists a design of visible-wedge line blind valve, which provides a full bore opening or positive blinding inside a seat in a three quarter enclosed body. No line residue can spill when wedges are being changed and no line movement is necessary when spectacles are changed.



4.6 KNIFE GATE VALVE

These are single seated valves used for slurry services. They are covered under MSS-SP-Standards. Being single seated valve, it can be used for only unidirectional operations.

In its simplest form, the sliding valves consist of two stationary steel plates each with large holes drilled through it. A third plate slides between stationary plates to perform valve operations. When open this design offers no resistance to flow. The ability of the gate to slip through the plate effortlessly is the success of this design. The edge of the gate is shaped to shear the solids and the elastomer seal is used to keep the solids away from entering the space between the plates when open and to clean the sliding blade when it retracts.

VALVE INSPECTION AND TEST

If specified in the purchase order, the valves shall be inspected by the Purchaser's inspecting authority at the place of manufacture before despatch. If not, the manufacturer shall supply a certificate stating that the valve and valve parts comply in all respects with the relevant standards and regulatory codes.

Unless additional inspection is specified in the purchase order, inspection by the Purchaser shall be limited to the following.

1.0 Visual examination of any finished components at the assembly stage,

2.0 Visual and dimensional check of the finished valve,

3.0 Witnessing of the required and specified optional pressure tests.


— Blind

Fig-8

generally kept after valves to avoid passing of fluid

The regulatory codes referred for these tests are either;

1. API - 598 - Valve inspection and test.

2. API - 607 - Fire safe testing of soft-seated valves.

3. BS - 6755 - Testing of Valves.

Pressure Tests

Pressure tests, unless otherwise specified, shall be carried out on each valve as follows;

1. Shell - hydrostatic (inclusive of body, bonnet, stuffing box and cover plate)

For this test, disks, wedges and plugs shall be in open position and ball in the half open position. Check valves shall have pressure applied to the upstream side. When valves with stuffing box are tested, the back seat shall be tested for leakage with valve in fully open position and stem packing removed or untightened.

2. Seat - hydrostatic

For gate valves, the test is to be conducted to each side of disk. For globe valves, pressure is to be applied under the disk. For check valves, it is on the downstream side. For ball valves and plug valves, it is to each side of ball/plug.

3. Seat - Pneumatic

All gate, globe, plug and ball valves are air seat tested at a minimum of 80-psig/100 psig with liquid on the side, which is not under pressure. This test is not required for check valves.

For valves used for vacuum service, a low-pressure air test on the seat shall also be carried out.

Hydrostatic Test Pressures (in bar)

PRESSURE CLASS (LBS)	SHELL	SEAT (NOTE # 1)
150	30	21
300	76	54
400	100	74
600	150	110
900	225	165
1500	373	274
2500	621	456
800	210	152

Note # 1. The hydrostatic seat test pressure of soft-seated valves shall not exceed the body rating or the seat rating whichever is lesser (refer regulatory code for the valves).

For hydrostatic tests, the test fluid shall be water at ambient temperature unless the use of another liquid is agreed. The use of high chloride - containing water should be avoided. The water may contain water-soluble oil or a rust inhibitor.

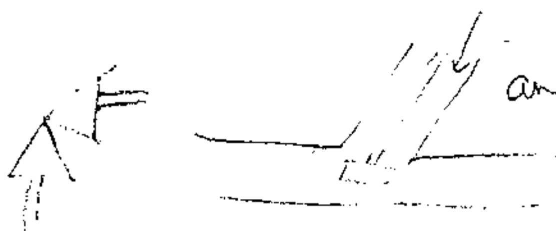
Fire safe test is a destructive test and carried out only in exceptional cases when specified. The arrangement and test shall be as per API 607 or BS 6755.

****

Regulatory Standard for Gate valves

1.	API-600/ISO-10434	Steel Gate Valves (BS EN ISO 10434)
2.	API-602 /ISO 15761	Steel Gate, Globe & check Valves DN100 & smaller (BSEN ISO 15761)
3.	API — 603	Corrosion Resistant Gate Valves
4.	ASME B 16.34	Steel Valves — Flanged and BW ends
5.	ASME 16.10 (EN 558-2)	Face to Face and End to End dimensions of Valves
6.	API 6D/ISO 14313	Pipeline valves
7.	BS 5 150	C I Wedge Gate Valves
8.	BS 5151	C I Parallel Slide Gate Valves
9.	AWWA C-500	C I Gate Valves for Water and Sewerage services
10.	AWWA C-509	Resilient Seated Gate Valves for Water and Sewerage services
11.	MSS-SP-70	C I Gate Valves
12.	IS 14846	C I Sluice Valves
13.	IS 778	Copper Alloy Gate Valves
14.	IS 10611	Steel Gate Valves for Petroleum Industries

Test pressure = design pressure X 1.5 x $\frac{\text{AS at amb. conditions}}{\text{AS at design temp.}}$



angle control valve

angle control valve

(for high pressure applications)

Regulatory Standard for Ball valves

1.	ISO 17292	Steel Ball Valves for Petroleum Industries (800#, 150#,300#,600#)
2.	BS5 1 59	C I Ball Valves for General Purposes
3.	MSS-SP-72	Ball Valves with Flanged and BW ends for General Purposes
4.	MSS-SP-122	Plastic Industrial Ball Valves
5.	AWWA C-507	Ball Valves 6' — 48"
6.	API6D/1S014313	Pipeline Valves
7.	ASMEBI6.34	Steel valves Flanged & BW ends
8.	ASMEB16.10 (EN 558-2)	Steel Ball Valves for Petroleum Industry

Regulatory Standard for Plug valves

1.	API-6D/ISO 14313	Pipeline Valves
2.	MSS-SP-78	Cast Iron Plug Valves
3.	BS 5158	C I and C S Plug Valves
4.	MSS-SP-108	Resilient Seated C I Eccentric Plug Valves
5.	API 599	Metal Plug Valves
6.	BS 5353	Steel Plug Valves

Regulatory Standard for Diaphragm valves

1.	MSS-SP-88	Diaphragm Type Valves
2.	BS EN 13397	Industrial diaphragm valves made of metallic material
3.	IS 11791	Diaphragm Type Valves for General Purposes

Regulatory Standard for Butterfly valves

1.	API 609	Butterfly Valves
2.	AWWA C-504	Rubber Seated Butterfly Valves
3.	MSS — SP - 67	Butterfly Valves
4.	MSS — SP - 68	High Pressure Butterfly Valves with Offset Design
5.	BS 5155	C I Butterfly Valves
6.	IS 13095	Butterfly Valves
7.	ASME 16.34	

Regulatory Standard for Globe valves

1.	BS 1873	Steel Globe Valves for Petroleum Industry
2.	API 602/ISO 15761	Steel Gate, Globe & check Valves DN100 & smaller (BSEN ISO 15761)
3.	BS 5152	Cast Iron Globe Valves
4.	MSS-SP-85	Cast Iron Globe Valves
5.	ASME B 16.34	
6.	IS 778	Copper Alloy Globe Valves
7.	IS 10605	Steel Globe Valves for Petroleum Industry

Regulatory Standard for Non-Return valves

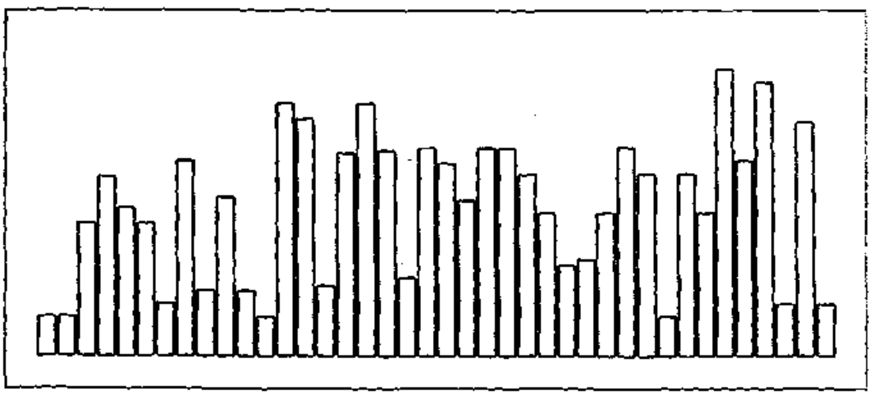
1.	BS 1868	Steel Check Valves for Petroleum Industry
2.	API 602/ISO 15761	Steel Gate, Globe & check Valves DN100 & smaller (BSEN ISO 15761)
3.	BS 5153	C I Check Valves for General Purposes
4.	IS 10989	Cast/Forged Steel Check Valves for Petroleum Industries
5.	IS 778	Copper Alloy Check Valves
6.	API 594	Wafer Type Check Valves
7.	IS 5312	C I Swing Check Valves
8.	API 6D/ISO 14313	Pipeline Valves

Certificate Course
on
PIPING ENGINEERING

June 12 - 25, 2006

BASICS OF PIPING DRAWINGS

T. N. Gopinath
Consultant



Organized by

Piping Engineering Cell
Computer Aided Design Centre
Indian Institute of Technology, Bombay
Powai, Mumbai - 400 076

PIPING ENGINEERING CELL

PIPING DRAWINGS

T. N. GOPINATH

THE BASICS

The drawings are always considered as the language of engineers. The machine drawings and the geometrical drawings are taught in the basic engineering curriculum. Piping Engineers derive basics from these to represent the pipeline routing on the drawing. There are two types of views used in the piping drawings:

a) Orthographic- Plans and Elevations
b) Pictorial - Isometric Views

Piping layout is developed in both plan view and elevation view and section / details are added for clarity wherever necessary. These drawings are called the General Arrangement of Piping. To represent a three plane piping in two dimensions of the paper, certain symbols are followed. Most commonly used symbols are in Table 1. Orthographic symbols are available in templates that are used for speeding up the manual drafting and also the symbol library for computer drafting.

In complex piping system, especially within the unit/plant building where orthographic views do not illustrate the details of design fully, pictorial view in isometric presentation is drawn for clarity. Specially printed isometric sheets are available with lines drawn vertically and at 30° clockwise and 30° counterclockwise respectively from the horizontal axis of the paper by the use of which 3D representation of the pipelines can be prepared.

1.0 PLAN AND ISOMETRIC PRESENTATION OF A PIPING SYSTEM

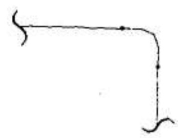
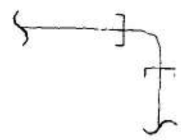
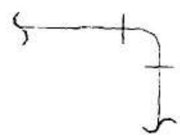
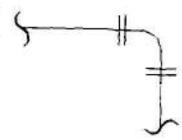
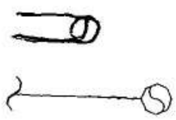
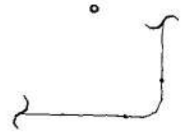
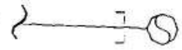
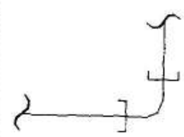
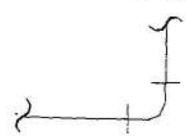
The purpose of drawing is to give detailed information so that the pipelines could be fabricated and erected to satisfy the process requirements. Prior to making the piping drawings the equipment layout drawings and plot plan are prepared and these drawings are used as the basis for developing the piping drawing. Sometimes preliminary piping study is made to fix the equipment co-ordinates. The other data required for the development of piping drawings are defined in the paper on "Equipment and Piping Layout".

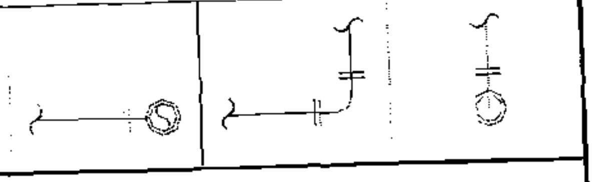
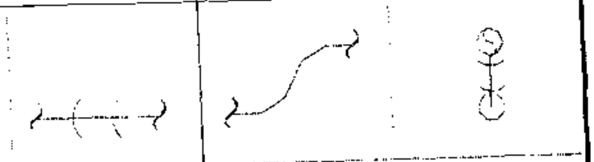
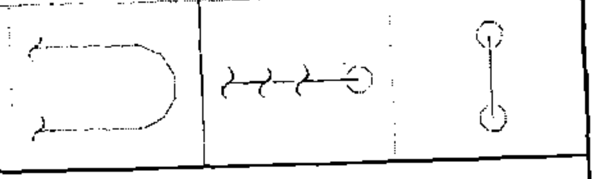
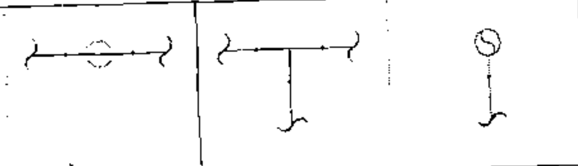
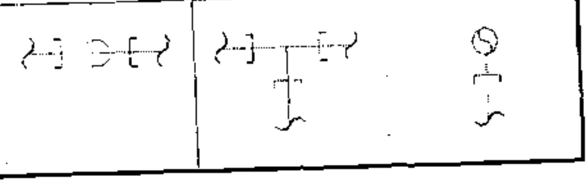
For presentation of unit piping layout the scale adopted usually are 1:25 or 1:33 1/3 and 1:100 for the pipe rack. There are different sizes of drawing sheets available for the preparation of the drawings. The Indian Standard IS10711 standardises the drawing sheets as below:

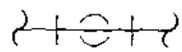
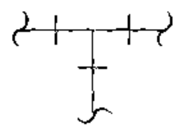
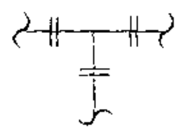
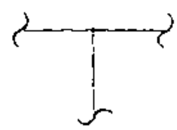
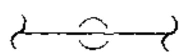
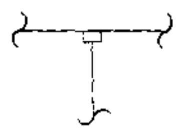
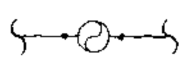
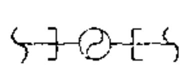
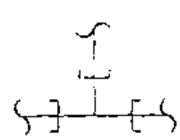
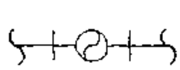
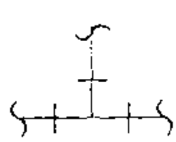
SIZE		OVERALL DIMENSIONS in mm (Untrimmed)
A_0	-	841 x 1189
A_1	-	594 x 841
A_2	-	420 x 594
A_3	-	297 x 420
A_4	-	210 x 297

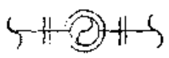
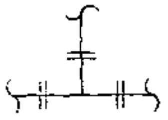
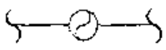
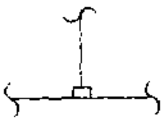
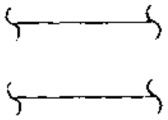
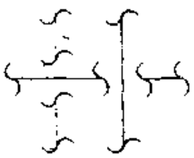
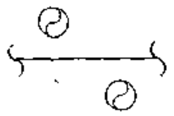
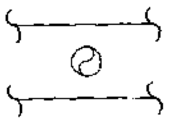
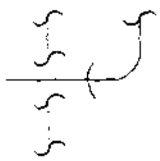
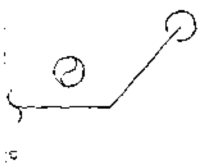
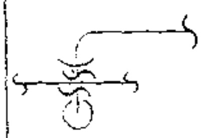
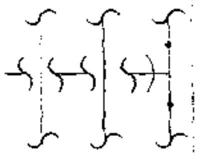
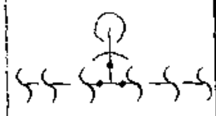
TABLE- I

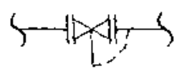
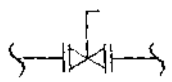
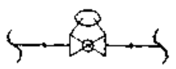
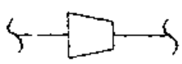
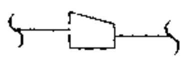
PIPING SYMBOLS

Sr. No.	DESCRIPTION	PLAN	END VIEW 1	END VIEW 2
1.0	CHANGE OF DIRECTION AT 90°			
1.1	DOWNWARD BENDING			
1.1.1	BW elbow			
1.1.2	SW elbow			
1.1.3	Scrd elbow			
1.1.4	Flgd elbow			
1.1	UPWARD BENDING			
1.2.1	BW elbow			
1.2.2	SW elbow			
1.2.3	Scrd elbow			

Sr. No.	DESCRIPTION	PLAN	END VIEW 1	END VIEW 2
1.0	CHANGE OF DIRECTION AT 90°			
	1.2 UPWARD BENDING			
	1.2.4 Flgd elbow			
2.0	CHANGE OF DIRECTION AT OTHER ANGLES			
	2.1 DOWNWARD BENDING			
	2.2 UPWARD BENDING			
	2.3 180° Returns			
3.0	BRANCHING			
	3.1 DOWNWARD			
	3.1.1 BW Tee			
	3.1.2 SW Tee			

Sr. No.			DESCRIPTION	PLAN	END VIEW 1	END VIEW 2
3.0	BRANCHING					
	3.1	DOWNWARD				
		3.1.3	Scrd Tee			
		3.1.4	Flgd Tee			
		3.1.5	Stub connection			
		3.1.6	Half Coupling			
	3.2	UPWARD				
		3.2.1	BW Tee			
		3.2.2	SW Tee			
		3.2.3	Scrd Tee			

Sr. No.	DESCRIPTION	PLAN	END VIEW 1	END VIEW 2
3.0	BRANCHING			
3.2	UPWARD			
3.2.4	Flgd Tee			
3.2.5	Stub connection			
3.2.6	Half Coupling			
4.0	PARALLEL LINES			
5.0	CROSS LINES			
6.0	ROLLED ELBOW			
7.0	ROLLED TEE			

Sr. No.	DESCRIPTION	PLAN	END VIEW 1	END VIEW 2
8.0	VALVES			
	8.1 Hand Wheel Operated Flgd Valve with vertical hand wheel			
	8.2 Lever operated Valve			
	8.3 Hand wheel operated BW Valve with rolled hand wheel			
9.0	CONCENTRIC REDUCER	a x b 		
10	ECCENTRIC REDUCER	 FSU/ FSD		

Piping General Arrangement is normally drawn on A_0 size sheet. If the area to be covered is small then A_1 size sheet is also used. Piping group produces a 'KEY PLAN', the plot plan on a small scale (1:500, 1:750 or 1:1000), which can be accommodated on an A_0 size drawing sheet as per scale and dividing the area of the site into smaller areas identified by key letters or numbers. This is added to the piping drawing for reference purposes. The subject area of the particular drawing is hatched or shaded.

The dimensional details of the title block are specified under the Indian Standard IS11665. The drawing sheet is divided along the length and the breadth in equal spaces. The longitudinal blocks are identified by alphabets and those along the breadth numerically. These co-ordinates are used to locate the area on the drawing. The direction of the north is taken either towards right or left on top of the drawing sheet. This direction is kept constant in all the areas covered in the plant, so also is the scale of the drawing.

2.0 HOW TO START THE PIPING GA ?

2.1 Obtain the drawings numbers and fill in the title block, with the drawing number and title at the bottom right hand corner of the sheet.

2.2 Place the north arrow at the top left/right hand corner of the sheet to indicate plant north.

2.3 Do not plan drawing in the area above the title block of drawing, as this is allotted for general notes, number and title of reference drawings, brief description of changes during revision and the bill of materials wherever applicable.

2.4 Process equipment and piping have priority on the Piping GA. The piping drawings are started after fixing positions of the equipments.

2.5 Inplant piping drawings are drawn to 1:33 1/3 scale and Piperack piping plan to 1:100 scale with junction details enlarged if necessary.

2.6 Equipment layout is reproduced on the Piping GA to its scale and drawn on the reverse side in case of manual drafting. In case of CAD separate layer is used.. The major primary beams and secondary beams are also shown if area covered is indoor.

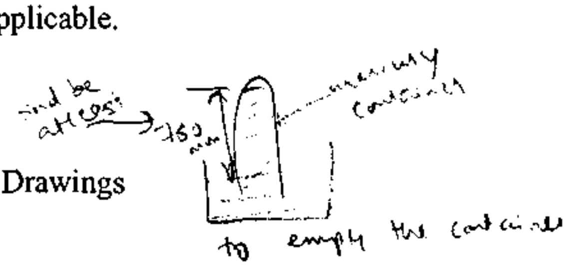
2.7 Pertinent background details which govern piping routing, such as floor drains, HVAC ducting, electrical and instrument cable trays, etc. are also drawn in faint on the reverse.

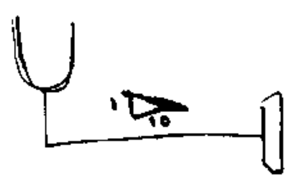
2.8 Utility stations are also established so that the most convenient utility header routing can be carried out.

3.0 DEVELOPMENT OF PIPING GENERAL ARRANGEMENT DRAWING

3.1 The piping drawings should be developed in such a way that all the process requirements are met with.

3.2 It is not always possible for the piping drawing to follow exactly the logical arrangement of the P & IDs. Sometimes lines must be routed with different junction sequence and line numbers and subsequently the list may be changed.





3.3 Performance and economics have to be considered in parallel while deciding the routing.

3.4 Piping is represented by single lines up to a size of 150NB and double lines for sizes 200NB and above. This is to save the time of drafting and to avoid confusion.

3.5 In single line representation only the centre line of the pipeline is drawn using solid line and in double line representation the actual size to scale is drawn with centre line marked in chain-dotted lines.

3.6 Line numbers are shown against each line exactly in the same way as represented in the P&I diagrams.

3.7 The change in specification should be shown in line with P&I diagram. This change is usually indicated immediately to the downstream of the valve, flange or equipment.

3.8 Valves should be drawn to scale with identification number from the P&ID marked thereon.

3.9 Draw valve hand wheels to scale with stem fully extended. If it is lever operated, then the movement of handle position should be marked.

3.10 If a valve is chain operated, note the distance of the chain from the operating floor.

3.11 Show location of each instrument connection with encircled instrument number taken from P&ID.

3.12 Similar arrangement shall be shown as typical detail or covered in a separate company standard as Instrument Hook-up drawings.

3.13 Draw plan view of each floor of the plant and these views should indicate how the layout will look like between floors as seen from top.

3.14 Each line should be identified by line number and should also show the insulation, tracing requirements, etc.

3.15 Lines, if required, shall be broken to show the required details of hidden lines without drawing other views.

3.16 Do not draw details that can be covered by a note.

3.17 Draw plan to a larger scale for any part needing more details and identify it as "Detail 'A'", etc.

3.18 Draw part isometric sketches or part elevations to clarify complex piping or piping hidden in the plan view.

3.19 Full sections through the plant may be avoided if isometric drawings are drawn for the lines. Part sections where required shall be shown to clear the hidden details in plan.

3.20 Sections in the plan views are identified by numbers say 1-1, 2-2, etc. and details by alphabets, e.g. "Detail 'A'".

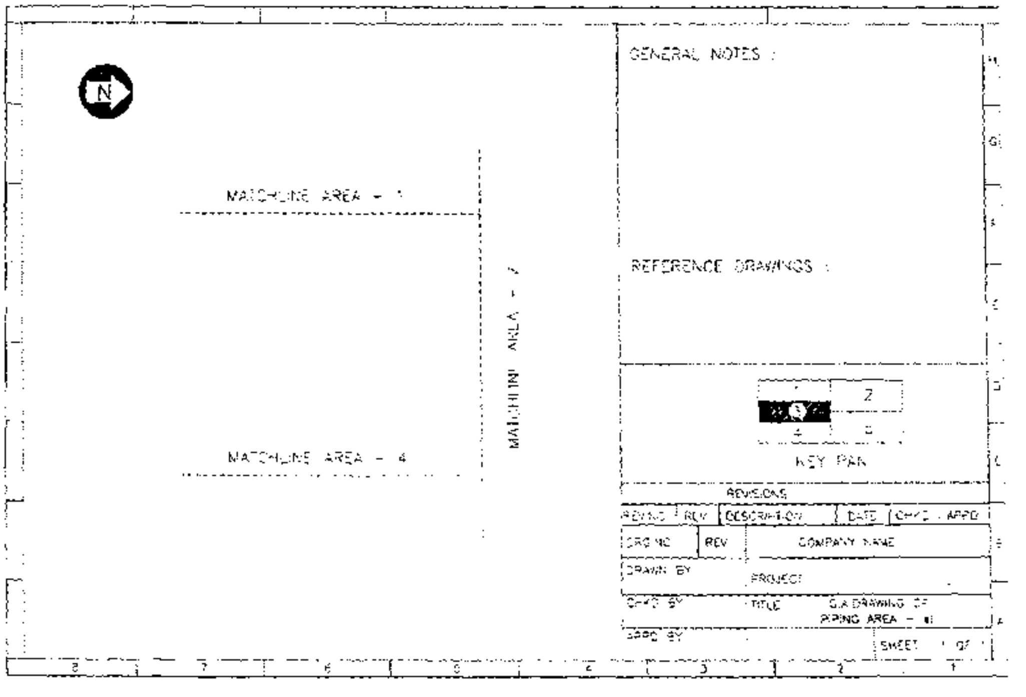


FIG. 1 : TYPICAL GENERAL ARRANGEMENT OF PIPING

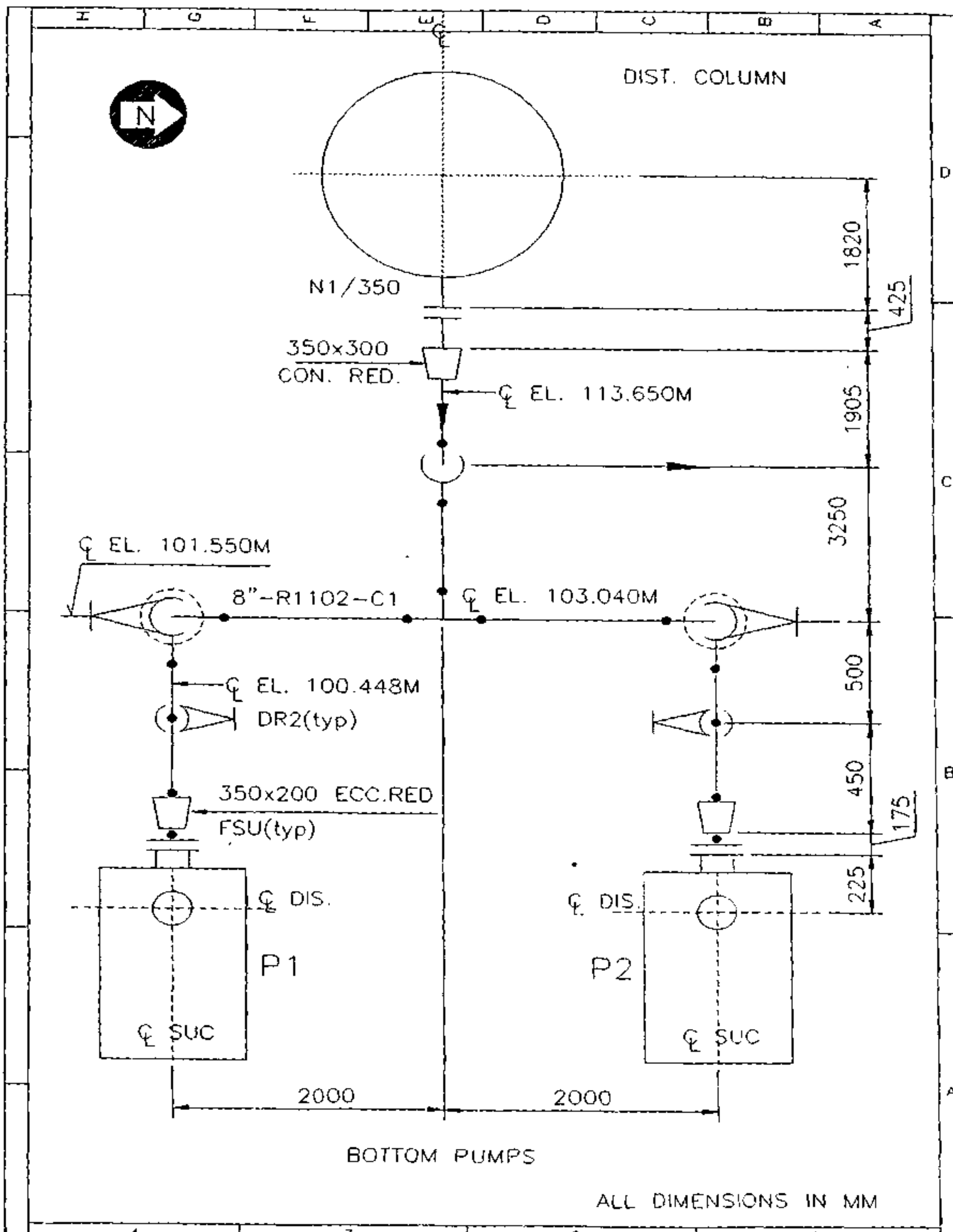


FIG. 2 : TYPICAL GENERAL ARRANGEMENT OF PIPING

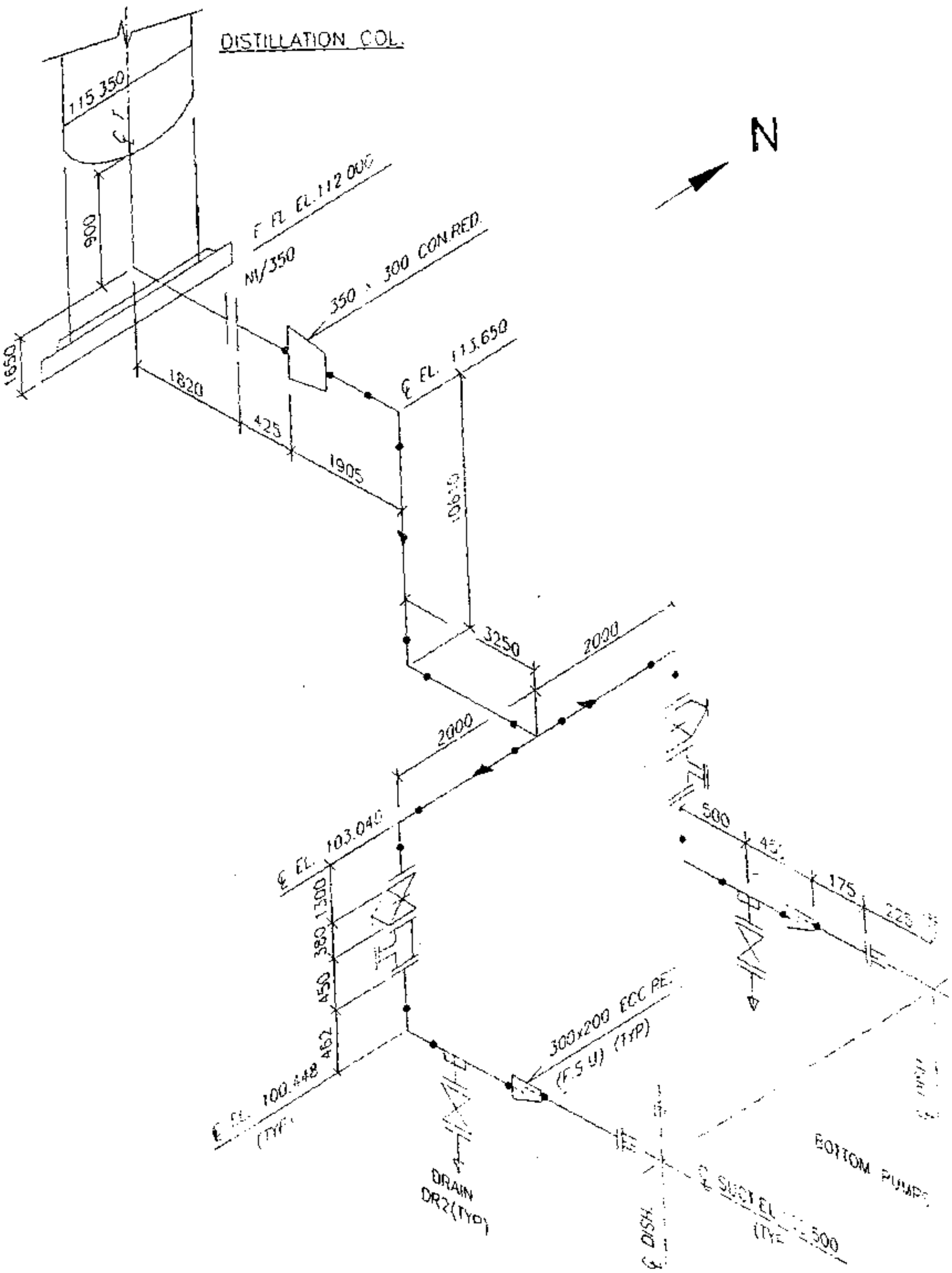


FIG. 3 : TYPICAL PIPING ISOMETRIC DRAWING

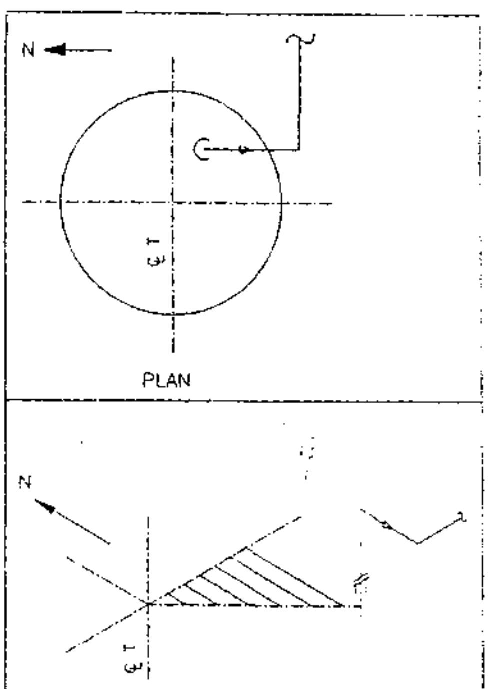
4.0 ISOMETRIC DRAWINGS OR ISOs

Piping isometrics are three dimensional representation or piping on two dimensional of drawing sheet. An isometric drawing covers a complete line as per the line list connecting one piece of equipment to another. It should show all information necessary for the fabrication and erection.

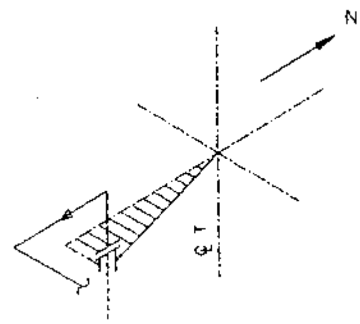
Isos are not drawn to scale but should be proportional for easy understanding. Dimensions are given relative to centreline of piping.

Isometric drawing should also include the following information:

4.1 Plant North - The direction should be so selected as to facilitate easy checking of GA with Iso



PROPER NORTH ORIENTATION



IMPROPER NORTH ORIENTATION

4.2 Dimensions and angles.

4.3 Reference number of P & IDs, GA Drawings, line numbers, direction of flow, insulation and tracing.

4.4 Equipment location and equipment identification.

4.5 Give nozzle identification on the connected equipment.

4.6 Give the details of flange on the equipment if the specification is different from the connecting piping.

4.7 Size and type of every valve/ Direction of operation.

4.8 Size and number of control valve.

4.9 Location, orientation and number of each equipment.

4.10 Field weld - preferred in all directions to take care of site variations. Can also be covered with a general note.

4.11 Location of high point vents and low point drains. Covered with a standard arrangement note.

4.12 Bill of Material.

4.13 Requirements of stress relieving, seal welding, pickling, coating, etc.

5.0 SPOOLS

When the piping is shop fabricated, the isometric drawings are developed further to create spool drawings. A spool is an assembly of fittings, flanges and pipes that may be prefabricated. It does not include bolts, gaskets, valves or instruments. A spool sheet is an orthographic drawing of a spool drawn either from piping GA or from an iso sheet. Each spool sheet shows only one type of spool and

5.1 Instructs welder to fabricate the spool

5.2 Lists the cut lengths of pipe, fittings and flanges etc. needed to make the spool

5.3 Gives material of construction and any special treatment of finished piping

5.4 Indicates how many spools of the same type are required

Spool numbers are given to make the identification easy. Iso sheets are identified with line number it represents. Both the spool and the spool sheet can be identified by a number or letter using the iso sheet number as prefix.

Straight run pipes over 6 m are usually not included in a spool, as such lengths may be welded in the system during erection in the field. The size of a spool is limited by the available means of transport.

As a general practice Carbon Steel piping 40NB and below are 'field fabricated'. All Alloy Steel and Carbon Steel spools 50 NB and above are normally 'shop fabricated'. Large diameter piping, being more difficult to handle, more economically produced in workshop.

6.0 DIMENSIONING OF DRAWINGS

6.1 Sufficient dimensions to be given for positioning equipment and for erecting piping.

6.2 Duplicating dimensions in different views should be avoided, as this may lead to errors if changes are made. Reserve horizontal dimensions for the plan view.

6.3 If single pipe is to be positioned or a pipe connected to nozzle is to be indicated, then show the centre line elevation and mark as ℄.

6.4 If several pipes are sharing a common support, show elevation of Bottom of Pipes and mark as BOP EL. This is more applicable to non-insulated lines.

6.5 In case of several pipes on a pipe rack, show the "Top of Support" elevation and mark as TOS EL.

6.6 In case of buried pipelines in trench, show elevation of bottom of pipes.

6.7 In case of drains and sewers, the Invert Elevation of the inside of the pipe is marked as IE.

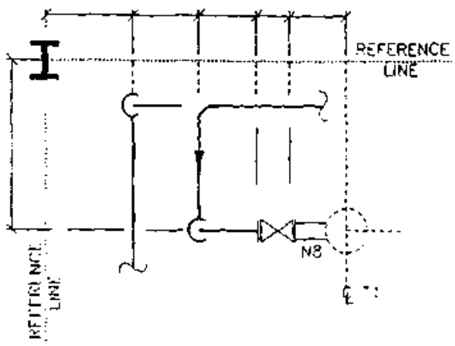
6.8 Centre lines of the equipment and pipelines shall be located with reference to the building column centre lines or the co-ordinates which can be considered as a reference base.

6.9 The distance between the lines shall be dimensioned centre line to centre line.

6.10 The horizontal nozzles on the equipment shall be located from

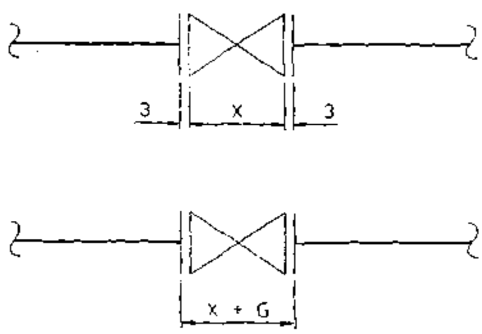
centre to flange face in plan. For vertical nozzles show Face of Flange elevation (FOF).

6.11 For valves, instruments and non - standard equipments, show the dimensions from flange face to flange face.



6.12 Flanged valves are located with dimension to flange faces. Non-flanged valves are dimensioned to their centres or stems.

6.13 For flanged joints show a small gap between dimension lines to indicate gasket. Flanged joints can also be shown without gasket but cover the same with a general note and include gasket thickness in the valve or equipment dimensions.

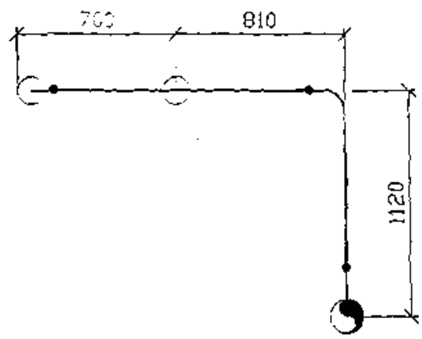


6.14 For Finished Floor (FF) the elevation shall be the high point of the floor.

6.15 For foundation the Top of Grout (TOG) elevation is shown.

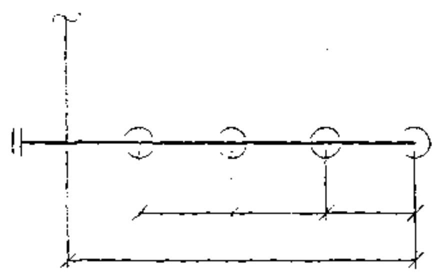
6.16 Show dimensions outside the drawn view - do not cut pictures.

6.17 Draw dimension line unbroken with fine line. Write dimension just above the horizontal line. For vertical lines write sideways.

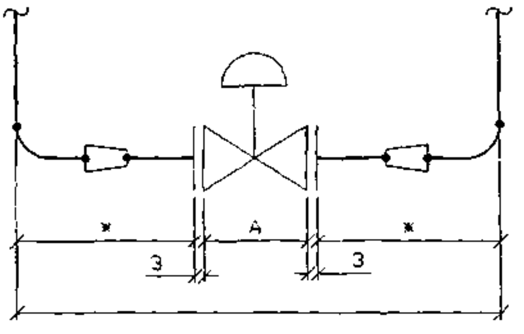


6.18 The dimension lines can be terminated with arrow heads or oblique dashes.

6.19 If series of dimension is to be shown, string them together. Show overall dimension of the string of dimensions. Avoid one of the break-up dimensions to omit repetition and error during changes.



6.20 Do not omit significant dimension other than fitting make up.



6.21 For field run piping, give only those dimensions which are necessary to route piping clear of equipments and other obstructions. Locate only those items which are important to the process.

6.22 Underline out of scale dimensions or mark as NTS.

6.23 Do not terminate dimensions at screwed or welded joints.

7.0 CHECKING OF PIPING DRAWINGS

Checking shall be done only on the print or the check plot of the drawings and by coloured pencils/pens.

A. Corrected areas and dimensions are marked yellow.

B. Areas and dimensions which are to be deleted are marked green.

C. Areas to be corrected/incorporated on the drawing are marked in red.

The new print after correction is "back checked" for incorporation.

Points to be checked on the piping drawing includes:

7.1 Title of the drawing.

7.2 Orientation - North arrow against plot plan.

7.3 Inclusion of graphic scale (if drawings is to be reduced).

7.4 Co-ordinates of equipments against equipment layout.

7.5 Equipment numbers and their appearance on the piping drawing.

7.6 Correct identification on all lines in all views.

7.7 Line specification changes.

7.8 Reference drawing numbers and files.

7.9 Correctness of all dimensions.

7.10 Whether representation is correctly made in line with the standard symbols or not.

7.11 Location and identification of all instruments. Requirements of upstream / downstream straight lengths.

7.12 Insulation requirements as per P&IDs.

7.13 Piping arrangement against P&ID requirements such as gravity flow, seals, etc.

7.14 Possible interference.

7.15 Correctness of scale in case of General Arrangement Drawings.

7.16 Whether all stress analysis requirements are met or not.

PIPING ENGINEERING CELL

7.17 Adequacy of clearance from civil structures, electrical apparatus and instrument consoles.

7.18 Floor and wall openings.

7.19 Accessibility of operation and maintenance space and provision of drop out and handling areas.

7.20 Foundation drawings and vendor equipment requirements.

7.21 Details and section identification match.

7.22 "Matchline" provision and accuracy.

7.23 Presence of signatures and dates.

7.24 Accuracy of BOM in Isometrics.

7.25 Number of issues and revision.

****

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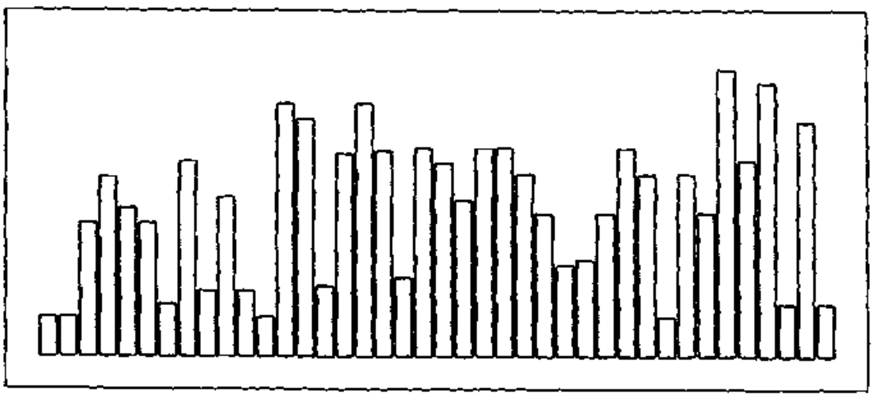
June 12 - 25, 2006

PLOT PLAN

T. N. Gopinath
Consultant



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Computer Aided Design Centre
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Powai, Mumbai - 400 076

PLOT PLAN

T. N. GOPINATH

The selection of a project site is a major corporate decision considering the various factors that make the plant technologically and economically viable. The industrial policy of the government is a major factor to be considered. Availability of the various supporting facilities such as power, water, effluent disposal, manpower etc. have also to be taken into consideration along with the size and nature of the industry. Once the site selection is done, the next activity is to develop the plot plan.

Plot plan is the master plan locating each unit/facility within the plot boundary for a process industry such as

i) Refinery
ii) Chemical / Agro Chemical / Petro-Chemical/ Organic Chemical / Inorganic Chemical
iii) Fertilizer
iv) Pharmaceutical
v) Metallurgical
vi) Power Generation

The development of plot plan is a much involved job. While locating the various units/facilities within the plot, consideration shall be given for the operation, maintenance, safety aspects related to the plant and that of the neighbour, fire hazards, location of power and water supply, expansion facilities, man-material movements, etc. in a balanced manner.

Before the activity of development of the plot plan starts, there are a lot of data, related to all disciplines of engineering, to be collected and analysed and/or made use of. Data to be collected before starting can be classified as follows.

1.0 BASIC DATA

1.1 Civil

1.1.1 PLANE TABLE SURVEY MAP

This document shows the extent of the plot with the overall area identifying all the existing constraints such as transmission lines, structures, ponds, place of worship if any etc. The survey covers 20 to 30 metres beyond the extent of the specified plot to show the nearby roads, drains, etc. and broad features of the neighbouring plots. This will also show the geographical north direction. The location of power and water connections are also shown therein. (Ref. Fig. 1)

1.1.2 CONTOUR SURVEY MAP

Contour survey map shows the levels of the plot with respect to the mean sea level. These levels are taken at 10 M Grids so that the topography of the plot is well represented. This will establish the co-ordinates; normally North-South and East-West or it could be X-Y as well. The contour map will also show the benchmarks indicating the mean sea level (MSL) to establish the level of the plot. (Ref. Fig. 2)

1.1.3 SOIL SURVEY

Soil survey is conducted to establish the bearing capacity of the soil which will be required for the civil design. The nature of the soil is also tested to determine the expansive nature, corrosion properties etc.

1.2 Electrical

The data related to electrical will constitute the supply voltage levels, the voltage levels required within the plant and fault levels to establish the power distribution system, the location of supply point to decide the location of the receiving station and the requirement of the state electricity boards.

1.3 Non Plant Facilities

Necessary data is required to be collected, to arrive at the block size of the following facilities, before the work is started on the development of plot plan. The user department should be consulted to arrive at the requirement and the allocation should also be proportionate to the available plot area and the effective process area.

1.3.1 Administrative Block

1.3.2 Canteen

1.3.3 Workshop

1.3.4 R & D, QC Laboratory and Pilot Plant

1.3.5 Gate House / Time Office

1.3.6 Security Arrangements

1.3.7 Vehicle Parking

1.3.8 Medical Centre

1.3.8 Ware House

- i) Covered Area
- ii) Open Area
- iii) Solid Warehouse
- iv) Liquid Warehouse

1.3.10 Steel / Scrap Yard

1.3.11 Fire Station

1.3.12 Weigh Bridge

1.3.13 Staff Colony

1.4 Meteorological Data

This data is required to arrive at the location of the process area, utility area and the type of enclosure required for the building, basic design parameters, the drainage details, etc. The data includes

1.4.1 Minimum, maximum and normal temperatures during the year

1.4.2 Rainfall

1.4.3 Intensity and direction of the wind (wind rose)

1.4.4 Seismic zone (earth movement)

1.4.5 Wet and Dry Bulb temperatures. (For cooling tower design calculation)

1.4.6 Flood level

1.5 Process data

A lot of data related with the working of the process plant is also required to develop a plot plan. These are

1.5.1 Size / capacity of the process unit to work out the area to accommodate the same.

1.5.2 Knowledge on the type of plant, whether it is to be located indoors or outdoors or the extent of enclosure required.

 (Oil Industry Safety Directory- Safety Direction)

1.5.3 Sequence of process flow to locate the process unit in the proper manner.

1.5.4 Hazardous nature of the plant to keep proper inter unit distances and work out the fire water storage volume.

1.5.5 The overall operating philosophy of the plant such as
- i) Fully Automatic
- ii) Partially Automatic
- iii) Manual
- iv) Batch / Continuous

1.5.6 Raw material receipt and product despatch philosophy

1.5.7 Storage Philosophy. The requirement of above ground and/or underground storages, the nature of storage material etc. are required to decide on the block size.

1.5.8 Effluent plant capacity and discharge points, incineration requirements, etc.

1.5.9 Type of hazard to decide fire hydrant system.

1.5.10 Number of flares. (90 mt area around Flare shd be free)

1.6 Data on Utilities

The following data on utilities are required to be gathered to size and locate the various utilities.

1.6.1 Source and/or supply point of Raw water

1.6.2 Quality of water available

1.6.3 Water consumption for the process

1.6.4 Requirement of different types of utilities such as Steam, Air, Nitrogen, DM Water, Soft Water, Cooling Water, Chilled Water, Brine, etc.

1.6.5 Capacities and the grouping philosophy based on the nature of utilities.

1.7 Statutory Requirements

Based on the location, nature and type of the plant, the requirements by the statutory authorities are well spelt out. Knowledge and application of these are essential to develop the plot plan. These talk about the requirements of the Green belt, Floor area occupation, Floor space index, Width of the roads, Free area to be maintained along the plot boundary, Height and tread of the steps, Floor to floor distances, requirements of distances to be maintained between the units, requirements within the petroleum storages and gas storages, fire fighting requirements, height of chimneys, etc. These requirements are as per the norms set by

1.7.1 State Industrial Development Corporation (SIDC)

1.7.2 Central / State Environmental Pollution Control Boards (PCBS)

1.7.3 Factory Inspectorate

1.7.4 State Electricity Boards (SEB)

1.7.5 Chief Controller of Explosives (CCOE)

1.7.6 Static and Mobile Pressure Vessel Rules (SMPV)

1.7.7 Tariff Advisory Committee (TAC)

1.7.8 Aviation Laws

1.7.9 Chief Inspector of Boilers (CIB)

If column/equipment is near flare then put ladder on shadow side / opposite side of flare.

1.7.10 Oil Industry Safety Directorate (OISD)

1.7.11 Food and Drug Administration (FDA)

1.7.12 Ministry of Environment and Forest (MoEF)

1.8 Expansion Philosophy

The philosophy of expansion within the unit and additional units should be considered while developing a plot plan. The expansion could be segregated as near future expansion and far future expansion and both should reflect in the overall plot plan.

2.0 DEVELOPMENT OF PLOT PLAN

Based on the data collected as listed above, the following details shall be worked out so that these can be used for the development of plot plan.

2.1 Block dimensions of :

2.1.1 Process plants considering the expansion philosophy

2.1.2 Utilities based on the grouping philosophy and expansion requirements

2.1.3 Electrical receiving station and sub-station

2.1.4 Uncovered storage spaces

2.1.5 Solid ware houses

2.1.6 Non explosive chemical storages/ Explosive chemical storages as per classification

2.1.7 Petroleum Storage as per classification

2.1.8 Fire water storage requirements

2.1.9 Acid / Alkali storage

2.1.10 Steel and scrap yard

2.1.11 Raw material storage and treatment facilities

2.1.12 Contractor's shed

2.1.13 Effluent treatment & Incinerator plants

2.1.14 Flare stacks

2.1.15 Control room

2.1.16 Administrative buildings, workshop, canteen, laboratories, pilot plant etc.

2.2 Tentative details of Pipe rack/Sleepers

2.3 Inter unit distance based on the type and nature of the process.

2.4 Safety distances for the storages based on the relevant statutory regulations.

3.0 POINTS TO NOTE

3.1. Normally Construction is permitted maximum on 50% of the plot area with total built up area equal to area of the plot (i.e. F.S.I. = 1 (Depending upon the regulation governing the area and the type of industry)

3.2. Area reserved for tree plantation shall be 1/3 of the area occupied.

3.3. Parking space – 10% of the plot area

3.4. Water storage capacity – 24 hr. minimum.
The following basics can be used to estimate the water requirement.

3.4.1 Domestic water - 100 litres per person per day

3.4.2 Water requirement for Boiler - Steam rating x Working factor

3.4.3 Cooling tower – 1 1/4 % of capacity as drift and blow down losses

3.4.4 Washing - 10-15 litres per day per sq.ft. of floor area of the plant

3.4.5 Gardening - 5 litres per day per sq.ft. of garden area

3.5. Height of Boiler Chimney H (in m) = 14 $Q^{1/3}$ where Q is the quantity of SO2 generated in kg/hr.

4.0 STEPS TO BE CONSIDERED WHILE DEVELOPING THE PLOT PLAN

4.1 Study the contour map and establish the grade levels/terraces.

4.2 Establish the N-S and E-W (or X-Y) grids, the plant north in relation to geographical north.

4.3 Establish the free area along the plot boundary as per the statutory norms.

4.4 Work out the area requirements for the green belt, vehicle parking etc. as per the norms.

4.5 The process blocks shall be located in the sequential order of process flow so that material handling (solid/liquid) is minimum.

4.6 The blocks shall also be arranged considering prevailing wind direction so that flammable gases do not get carried to sources of ignition.

4.7 Storage tanks shall be grouped according to process classification.

4.8 Centralised control room shall be located in safe area close to process plant.

4.9 Two adjacent process units shall be located based on annual shut down philosophy so that hot work shall not affect the operation.

4.10 Process unit shall be located on higher ground away from the unwanted traffic.

4.11 Process units shall be serviced by peripheral roads for easy approach.

4.12 Utility block shall be kept at safe area close to process plants.

4.13 Electrical sub-stations shall be placed at the load centre to minimise cabling.

4.14 Receiving station shall be placed near the supply point.

4.15 Ware houses shall be located close to the material gate to avoid truck traffic within the process area.

4.16 Flares, Furnaces/Heaters, cooling towers, etc. shall be placed depending on the wind direction.

4.17 Provision of future expansion shall be considered.

4.18 Raw water storage shall be placed closer to water source. Fire and raw water tanks shall be located together.

*Cooling water motors → highest capacity motor

*utilities require a lot of power.

4.19 Fire stations shall be away from the hazardous area and nearer to main gate.

4.20 Effluent treatment plant shall be located away from the process and utility area on the downwind direction.

4.21 Workshop, contractor's shed, storage yard, etc. shall be at centralised location serviced by peripheral roads.

4.22 Two gates are preferred, one for the material entry with weigh bridge and the other one for man entry.

4.23 Administrative block, laboratories, etc. shall be located closer to the man entry gate.

4.24 Process unit can be separated within a fencing providing additional gate.

4.25 Consider recommendation from the statutory authorities for inter unit distances.

4.26 Residential colony shall be located away from the plant more closer to the city limits.

5.0 POINTS TO CONSIDER WHILE DEVELOPING THE EXPLOSIVE TANK FARM WHICH REQUIRES CCOE APPROVAL.

5.1 Layout of Liquid Storage

5.1.1 <u>CLASSIFICATION OF PETROLEUM PRODUCTS</u>

i) Class-A - Liquids, which have flash point less than 23°C

ii) Class-B - Liquids which have flash point 23°C and above but below 65°C

iii) Class-C - Liquids which have flash point 65°C and above but below 93°C

iv) Excluded Petroleum: Liquids, which have flash point above 93°C

5.1.2 <u>REGULATORY QUANTITY ABOVE WHICH LICENCE IS NECESSARY</u>

i) Petroleum Class A - 30 litres
In case of motor conveyance or stationary engines, capacity of fuel tank.

ii) Petroleum Class B - 2,500 litres
provided it is contained in a receptacle not exceeding 1,000 litres capacity

iii) Petroleum Class C - 45,000 litres

5.1.3 <u>LAYOUT CONSIDERATIONS FOR EXPLOSIVE TANK FARM</u>

i) Petroleum storage tanks shall be located in dyked enclosures with roads all around the enclosure.

ii) Dyked enclosure should be able to contain the complete contents of the largest tank in the tank farm in case of an emergency. Enclosure capacity shall be calculated after deducting the volume of the tanks (other than the largest tank) upto the height of enclosure. <u>A free board of 200 mm</u> shall be considered in fixing the height of the dyke.
In case of excluded petroleum the capacity of the dyked enclosure could

be based on spill containment and not containment on tank rupture.

iii) The height of tank enclosure dyke shall be at least 1 M and shall not be more than 2 M above average ground level inside. However, for excluded petroleum it can be 600 mm.

iv) Class A and/or Class B petroleum can be stored in the same dyked enclosure. When Class C is stored together, all safety stipulations applicable to Class A and Class B shall apply.

v) Excluded petroleum shall not be stored in the same dyke.

vi) Tanks shall be arranged in two rows so that each tank is approachable from the surround road.

vii) The tank height shall not exceed one and a half times the diameter of tank or 20 M whichever is less.

viii) Minimum distance between the tank shell and the inside of the dyke wall shall not be less than one half the height of the tank. Height is considered from bottom to the top curb angle.

ix) It is better that the corner of the bund should be rounded and not at right angle as it is difficult extinguish fire in a 90^0 angle corner because of the air compression effect.

x) There should be a a minimum of two access points on opposite sides of the bund to allow safe access/ escape in all wind directions.

xi) Distances to be observed around facilities in an installation shall be as per the relevant chart furnished in the Petroleum Rules. (Refer Fig. 3 & relevant Table in the Petroleum Rules)

5.2 Layout Of Gas Storage

5.2.1 Storage Vessels are not allowed below ground level.
They are to be installed above ground level.

5.2.2 Vessels shall be located in open.

5.2.3 Vessels are not to be installed above one another.

5.2.4 If vessels in the installation are more than one, the longitudinal axis of vessels should be parallel to each other.

5.2.5 Top surfaces of vessels are required to be made in one plane.

5.2.6 Vessels installed with their dished ends facing each other shall have screen walls in between them.

5.2.7 The distances to be observed between two vessels in one installation and distance from building or group of building or line of adjoining property are given in Table 1 & Table 2.

5.2.8 The area where vessels, pumping equipment, loading and unloading facilities and direct fired vaporizers are provided shall be enclosed by an Industrial Type Fence at least 2 M high along the perimeter of Safety Zone.

5.2.9 The minimum distances to be observed around installation shall be as per the guidelines in SMPV, which are reproduced in Table 1 & Table 2.

5.2.10 Not withstanding anything contained in the sub rules above, the storage of LPG can be placed underground or covered by earth in such a manner and subject to such conditions as may be specified by the notifications by the Central Government.

5.2.11 Above ground vessel for storage of corrosive flammable or toxic gas in liquefied state shall be provided with enclosure wall all around the ground. The minimum distance between vessel and enclosure wall shall be the diameter of the vessel or five meters whichever is less. Ground shall be graded to form a slope away from pumps, compressors or equipments. The height of the enclosure shall be 30 cm. on the upper side and gradually increasing to 60 cm. On the lower side at the end of which a shallow sump for collection of spilled liquid if any, shall be provided.

TABLE 1
Minimum Safety distances for flammable, corrosive & toxic gases

Sl. No.	Water capacity of Vessels (in litres)	Minimum distance from Building or Group of bldgs/line of adjoining property	Minimum distance between Pressure Vessels
i	Not above 2000	5 metres	1 metre
ii	Above 2,000 but not above 10,000	10 metres	1 metre
iii	Above 10,000 but not above 20,000	15 metres	1.5 metres
iv	Above 20,000 but not above 40,000	20 metres	2 metres
v	Above 40,000	30 metres	2 metres

TABLE 2
Minimum Safety distances for non-toxic gases

Sl. No.	Water capacity of Vessels (in litres)	Minimum distance from Building or Group of bldgs/line of adjoining property	Minimum distance between Pressure Vessels
i	Not above 2000	3 metres	1 metre
ii	Above 2,000 but not above 10,000	5 metres	1.5 metre
iii	Above 10,000 but not above 20,000	10 metres	2 metres
iv	Above 20,000	15 metres	Diameter of larger vessel

Note: The distances specified above may be reduced by the Chief Controller in cases where he is of the opinion that additional safety measures have been provided.

TABLE - 3

Minimum Clearances to be considered in a Process Unit

(As per OISD guidelines)

1.	Process units to flare	90M
2.	Storage tanks class A/B	0.5D or 15M for Class A/B, 6M for Class C
3.	Storage tank to vehicle unloading	15M - Class A/B 3M - Class C
4.	Vehicle unloading to boundary facing	15M - Class A/B 3M - Class C
5.	Storage tank periphery to boundary facing	15M - Class A/B 4.5M - Class C
6.	Electrical substation to process units	15M
7.	Head room over main refinery roads	7.6M
8.	Head room over main service roads	6M
9.	Head room over secondary roads	4.5M - for cranes 3.6M - for trucks

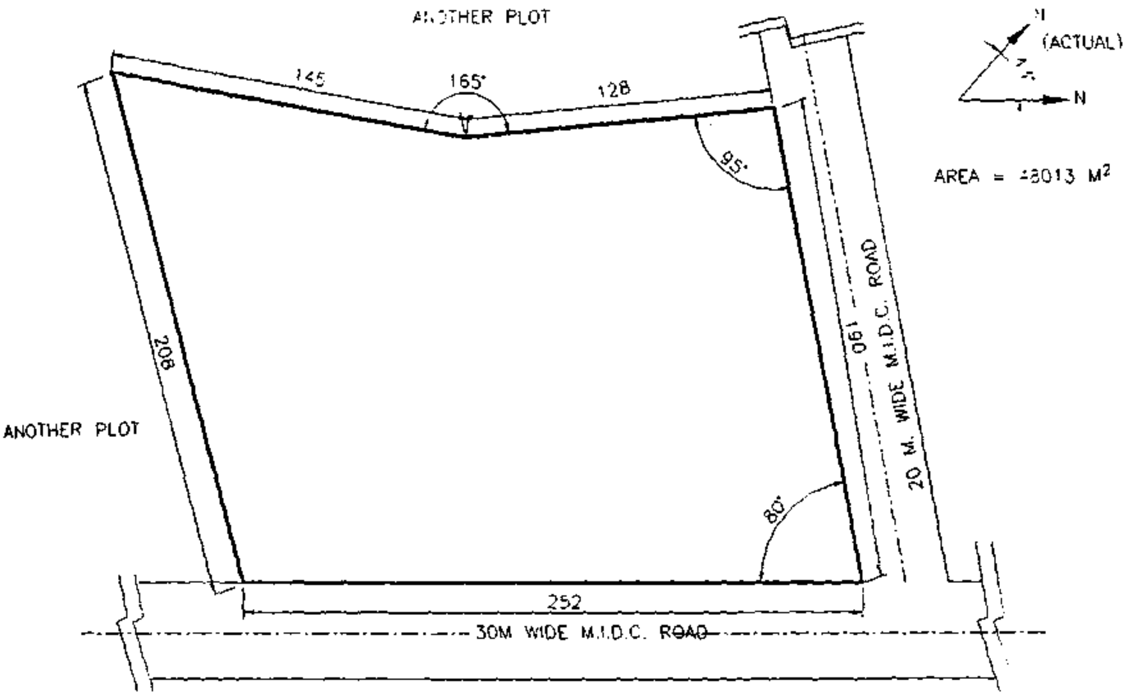

ANOTHER PLOT
145
165°
128
95°
208
190
ANOTHER PLOT
20 M. WIDE M.I.D.C. ROAD
80°
252
30M WIDE M.I.D.C. ROAD
N (ACTUAL)
N

Fig. 1

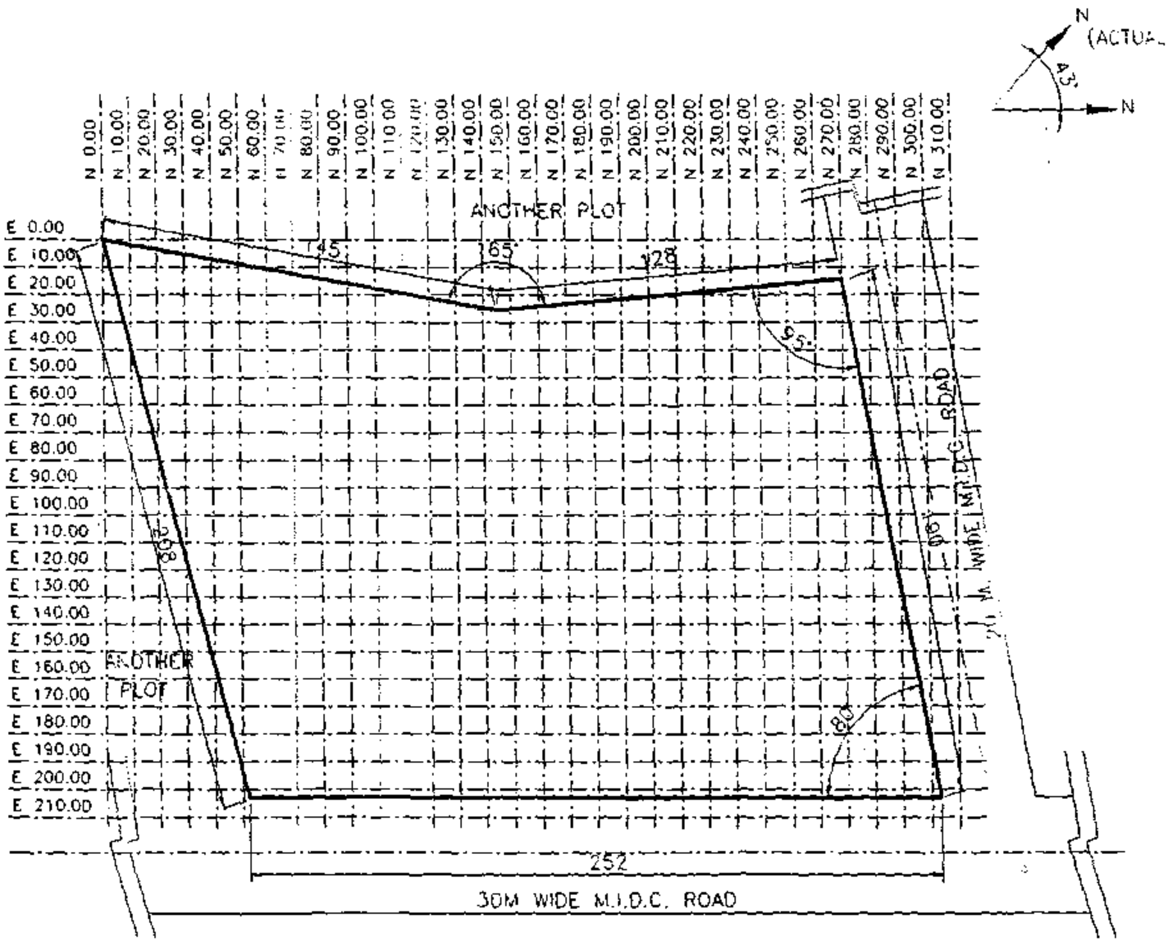

N (ACTUAL)
43°
N
N 0.00
N 10.00
N 20.00
N 30.00
N 40.00
N 50.00
N 60.00
N 80.00
N 90.00
N 100.00
N 110.00
N 130.00
N 140.00
N 150.00
N 160.00
N 170.00
N 180.00
N 190.00
N 200.00
N 210.00
N 220.00
N 230.00
N 240.00
N 250.00
N 260.00
N 270.00
N 280.00
N 290.00
N 300.00
N 310.00
ANOTHER PLOT
E 0.00
E 10.00
E 20.00
E 30.00
E 40.00
E 50.00
E 60.00
E 70.00
E 80.00
E 90.00
E 100.00
E 110.00
E 120.00
E 130.00
E 140.00
E 150.00
E 160.00
E 170.00
E 180.00
E 190.00
E 200.00
E 210.00
145
165°
128
95°
208
ANOTHER PLOT
ROAD
80°
252
30M WIDE M.I.D.C. ROAD

Fig. 2

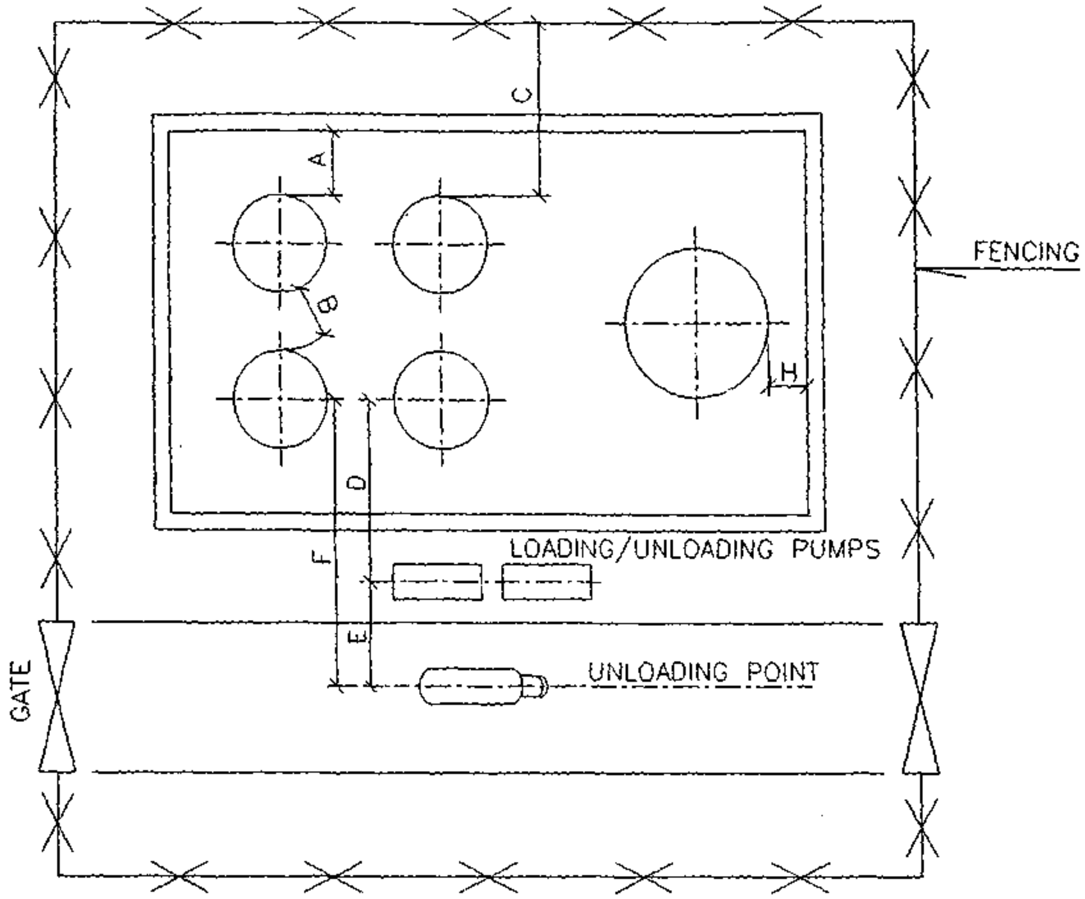
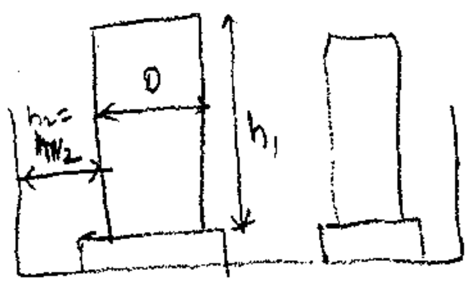


Fig. 3: EQUIPMENT LAYOUT - EXPLOSIVE TANK FARM



$h_2 = 0.8 - 1D$ (if D is considered)

$= 0.5h_1$ (if h_1 is major factor)

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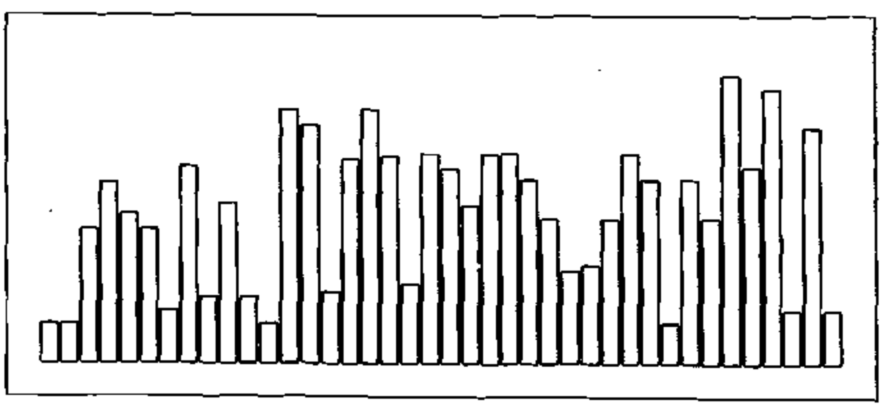
June 12 - 25, 2006

EQUIPMENT AND PIPING LAYOUT

T. N. Gopinath
Consultant



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EQUIPMENT AND PIPING LAYOUT

T. N. GOPINATH

1.0 INTRODUCTION

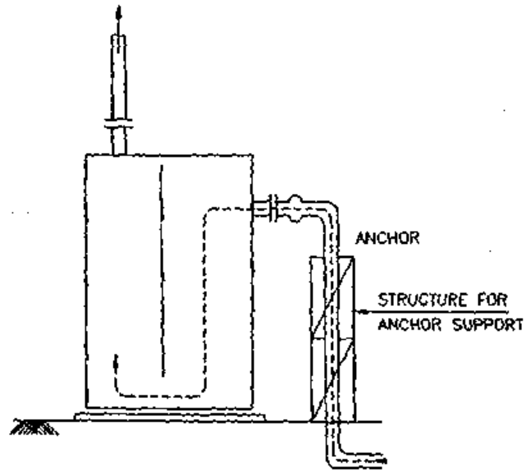
Equipment and Piping arrangements cannot be segregated to have different approaches since the requirements of equipment and piping layout design often overlap.

Equipment is arranged in the process flow sequence in plan and elevation and piping is laid to effect the process flow. It is very appropriate to say that the Equipment and Piping layout design is an ART and not a SCIENCE. There is not a single formula available for the design of Equipment and Piping layout. The equipment layout design can be as rational as the mathematics of fluid flow but with the language of projective geometry. Mathematics is abstract; geometry is visual. All engineering courses have mathematics; few have the subject of projective geometry but none has layout design. However, systematic methods and procedures can be developed from engineering principles, specifications, practical engineering knowhow, and just SIMPLE COMMON SENSE. All this should be coupled with the capacity to visualize the arrangement of equipment and piping three dimensionally. The design must take constructibility, economics, safety, quality and operation into account. All these should be achieved within the shortest schedule and will demonstrate the technical capacity along with creative talent and common sense approach to problem solving. Although the tools to achieve these goals have changed from pencil and paper to computer graphics, the responsibilities of the Piping Engineer remains the same.

During the planning stages, the Piping Engineer could meet with simple ideas that can effect substantial cost savings. Let us take a practical example to it.

In a chemical process industry, a waste heat boiler had to be installed at the exhaust of a diesel engine to recover the waste heat. The job was awarded to a Consulting organization on a turnkey basis. The design activities took the following sequence.

The process group worked out the required parameters did the process design of the heat exchanger and issued Process Data Sheet (PDS) to the Fabricated Equipment Group (FEG), who did the mechanical design and issued the drawing to the Piping Group. Piping Group located the heat exchanger and designed the inlet piping and also did the flexibility analysis of the same piping as it is subjected to high temperature. To save the equipment, an anchor was placed near the inlet nozzle with an expansion bellow. The data of the anchor loading was passed on the Structural Group, who designed a braced structure to take care of this. The total cost became prohibitive. At this point, a suggestion was made to turn the waste heat boiler upside down and thus eliminating the heavy structure. The final design turned out to be simple, cost effective and occupied less space. (Refer Fig. 1.1a, 1.1b)



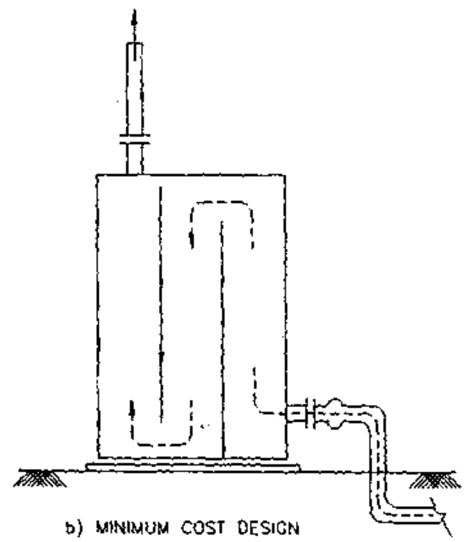


Fig. 1.1a, 1.1b

2.0 STEPS IN PLANT DESIGN

The mechanical design and development of the plant has three major steps viz.

2.1 Conceptual layout design
2.2 Equipment layout design
2.3 Piping layout design

These are not sharply divided areas. Though equipment arrangement can be made along with the piping layouts, it is normally dealt with separately in large plants.

The plant layout can be the biggest cost saver in chemical plant design next to the Process and Equipment design. Money wasted or saved can be substantial between alternate layouts. In addition to capital cost, the plant layout also influences the operating and maintenance cost. These are long-term benefits that affect profitability.

Incorrectly established plant layouts can have serious impact on safety and operability. If the layout does not have enough room, the plant will be overcrowded, unsafe and difficult to operate and maintain. On the other hand, an overly generous layout results in unnecessary high capital investment.

2.1 Conceptual Layout

Conceptual layout is a part of the basic engineering package. The design of it is a highly innovative activity.

In this, only the essential process design requirements are established, horizontal and vertical relationship of equipment is spelt out. Space allocation is there for all the basic plant requirements such as laboratories, offices, storage etc. Access for operation, maintenance and construction is provided for. Control room, motor control centre room etc. is also planned. The basic size of the building/structure is worked out. The resultant drawing is the conceptual layout. Normally small scales of the order of 1:200 or 1:100 are used to represent the same along with simplified presentation techniques. Plans along with necessary cross sections complete this drawing. Changes in the process, operating philosophy or equipment type and size can end up in substantial changes in the conceptual arrangement. Hence it is imperative that, being a basic document, proper thought should be given while generating the conceptual layout. A typical conceptual layout is illustrated in Fig. 2.1.1.

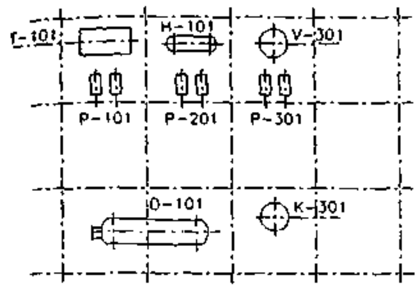


Fig. 2.1.1a PLAN GROUND FLOOR

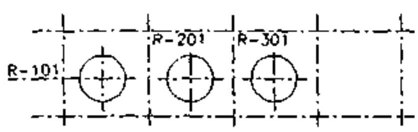


Fig. 2.1.1b PLAN MEZ. FLOOR

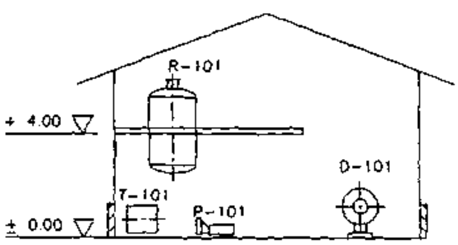


Fig. 2.1.1c SECTION

2.2 Equipment Layout:

Equipment layout is an extension of the conceptual layout in a more detailed manner. In the same way as the P & I diagrams are the basic documents of chemical engineering design, equipment layout is the basic document of mechanical engineering design. This is a composite mechanical engineering design, coordinating the design information to produce construction drawings.

Generally all equipment and facilities that need floor space are shown. Access, removal space, cleaning area, storage space and handling facilities are outlined. Good layout design minimizes the cost of operation and maintenance. Access is the most important feature to be considered Constructibility is another factor, which have equal importance. Equipment layout of large outdoor plant is sometimes referred as plot plan. This document is the basis for the development of construction drawings by all disciplines.

Information required for the preparation of the equipment layout is more extensive than those required for the concept layout design. The essential data required for the preparation of an Equipment Layout is as follows.

2.2.1 PROCESS FLOW DIAGRAMS (PFD) / PIPING AND INSTRUMENT DIAGRAMS (P &ID)

PFD / P&ID is the most important document referred to by a layout engineer. They show how each of the equipment is interconnected. P & ID indicates the information such as solid handling, gravity feed, line slopes, loop sizes, venting and draining requirements, special piping materials etc. which govern the equipment location to a great extent.

Utility flow diagrams show the individual service lines and utility headers.

2.2.2 PROJECT DESIGN DATA

Project Design Data includes the geographic location, proximity to roads and railways, topography and local codes and regulations. It also lists weather conditions such as rainfall records, seasonal temperature differences, wind directions, outlet point for drains etc. This data affects the design of storm water drains and requirement of enclosures. Further, the wind direction influences the location of cooling towers, furnaces, incinerators, stacks etc.

Grade elevation is usually referred to a datum such as +100.00 M or ± 0.00M and is referred to an absolute level in the Plot Plan. This is fixed at the ground floor of one of the major process plants and it is absolutely essential to establish a consistent elevation relationship between various facilities. The plants and major equipment are located with reference co-ordinates that are established in the Plot Plan.

The zero level '100' is mean see level. The level below that is

The soil characteristics decide the foundation depth and the required area to be occupied. This affects the equipment spacing to take care of foundation interference and layout of drains.

2.2.3 EQUIPMENT SIZES AND BUILDINGS

Equipment includes fabricated equipment such as Vessels, Heat Exchangers, Reactors, Tanks and proprietary equipment such as Pumps, Compressors, Furnaces, Filters etc. The layout and piping arrangement characteristics of each of these shall be dealt with subsequently. However certain general principles are followed in locating the same. The equipment is grouped to have the optimum location for minimum pipe run. Process flow sequence is followed to establish the functional performance of the same.

a) Inline Layout

Exchangers are placed next to towers. Thermosyphon reboilers, which have large diameter pipelines, are attached to towers as well. Towers are arranged with individual platforms or with a common platform for several in a line. Generally, surge drums, storage vessels, coolers, heaters are placed between distillation columns in the process flow sequence.

(see fig. in stress analysis chapter)

b) Similar equipment grouping

Operating or maintenance convenience and safety considerations can dictate the grouping of equipment. Grouped exchangers and lined up channel ends make possible the use of a common gantry crane moving on rails in the front for bundle removal. Columns can be lined up to have a common platform for manholes and valve operations. Reactors or agitated kettles can be grouped to have common operating levels and lifting beam for the agitators. Utility equipment is normally housed together for operating convenience. Process compressors under one roof are a frequent requirement. Pumps in a row can facilitate better rack layout, cable arrangement and maintenance accessibility

c) Functional equipment grouping

Cost of alloy steel/stainless steel piping initiates a compact arrangement of equipment. Bolting condensers on top of distillation column as a part of the same, stacking heat exchangers one above the other are some examples. Reactors are arranged in a row, which need crane or trolley for removing equipment internals and for material handling. Equipment containing acids or toxic materials is grouped and located within a paved and curbed area which will have the facility to drain out the effluent treatment plant or for neutralizing it. The building sizes in an outdoor plant include the MCC room, control room, laboratories and space required for personal facilities. These are worked out in co-ordination with Electrical, Instrument and Civil/Structural groups.

2.2.4 GUIDELINES – A NEW APPROACH

Layout guidelines for major equipment are based on minimum distance required between various types of equipment to meet the insurance regulations. They are designed to help prevent major catastrophes, especially when highly inflammable chemicals are involved. These guidelines, while useful, often include ambiguous entries such as "provide spacing based on access for operation and maintenance", "as required", and "not applicable" etc. Hence following these guidelines only could still result in crowded plant. The solution is to study each piece of equipment to determine the needs of operation and maintenance. This, however, is an

extremely time consuming activity, and is quite possible to be overlooked or not accounted for certain specific need. As a result, plant layout becomes a subjective topic, left to an experienced Piping Engineer.

A quantitative method is evolved by earlier Piping Engineers for evaluating the quality of a given layout in the early part of design. Two complimentary calculations meet these criteria. The results are then compared to those of plants in similar service – both good and bad layouts. Used in conjunction with established practices, this new approach can improve engineer's ability to evaluate the equipment layout.

a) Area Calculation

Make a scale drawing of the layout showing all the major equipment such as Reactors, Exchangers, Filters, Columns, Vessels, Pumps, and other devices. Measure the area actually occupied by each equipment and the total area of the plant. Now compare this value with that of the other plants of the same or similar service. Ask the operators and maintenance personnel of these plants whether the area is adequate, crowded or roomy. Thus compare the proposed layout with that of the existing plants.

b) Volume Calculation

As a second part of the evaluation, make a similar study by comparing the equipment volumes. Calculate the volume of each major piece of equipment with the total volume of the plant. Then compare the results with volume ratios of the existing plants in similar service.

Adjust the size of plant and the equipment layout, if necessary. The following guidelines and cautions are helpful in improving the accuracy comparisons.

i) Make comparison to as similar a plant as possible.
ii) Use similar assumptions in analyzing both existing facilities and new design.
iii) For outdoor installation, where volume has less relevance than in an enclosed structure, rely on the area comparison alone.
iv) For tank farm, general guide lines dictated for fire safety reasons or statutory requirements govern.

General guide lines for equipment minimum spacing shall be as given in the Table.

GUIDELINES FOR EQUIPMENT MINIMUM SPACING

Key : All distances in meters
N.A. = Not applicable
x = Provide spacing based on access for operation and maintenance.
y = As required

How to use this table:
For equipment pairs specifically listed, read vertically down from one item and horizontally across from the other. The intersecting square holds the spacing.
For equipment not paired, use the basic spacing, which is listed around margin.

Reference letter	Facility	Basic spacing	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W
A	Gas compressors	7.5	x																						
B	Compressor drivers, other than steam or motor	10.5	x	x																					
C	Central control houses	30	30	30	x																				
D	Unit control houses	15	15	15	x	x																			
E	Air fin coolers	3	7.5	10.5	30	15	x																		
F	Cooling towers	15 & 30	15	15	30	15	15	x																	
G	Drums	1.5	7.5	10.5	30	15	3	15	1.5																
H	Electrical unit load centers or unit substations	15	15	x	x	x	15	15	15	x															
I	Critical electrical switch racks	4.5	4.5	x	x	x	x	15	4.5	x	x														
J	Emergency valves for shutoff, isolation snuffing steam, water Tank etc.	7.5	7.5	7.5	x	x	7.5	N.A.	7.5	N.A.	N.A.	x													
K	Exchangers operating above 315°C	4.5	7.5	4.5	30	15	4.5	15	4.5	15	4.5	7.5	1												
L	Exchangers operating below 315°C	1.5	7.5	10.5	30	15	3	15	1.5	15	4.5	7.5	4.5	1											
M	Furnaces	15	22.5	7.5	30	15	15	30	15	15	7.5	7.5	7.5	15	y										
N	Pumps above 315°C	4.5	7.5	4.5	30	15	4.5	15	4.5	15	4.5	7.5	4.5	4.5	7.5	1.5									
O	Pumps handling flammable materials	3	7.5	10.5	30	15	3	15	3	15	4.5	7.5	4.5	3	15	4.5	1								
P	Equipment handling non-flammables	x	x	x	x	x	x	x	x	x	x	x	4.5	x	4.5	4.5	x	x							
Q	Reactors above 315°C (Externally insulated)	4.5	7.5	4.5	30	15	4.5	15	4.5	15	4.5	7.5	4.5	4.5	4.5	4.5	4.5	4.5	1.5						
R	Reactors above 315°C (Internally insulated)	[illegible]	7.5	4.5	30	15	7.5	15	7.5	15	4.5	7.5	7.5	7.5	7.5	7.5	7.5	4.5	7.5	1.5					
S	Onsite pressure storage vessels	NA	22.5	22.5	30	22.5	22.5	22.5	22.5	22.5	4.5	22.5	22.5	22.5	2.25	22.5	22.5	x	22.5	22.5	1 dia				
T	Main equipment structures	1.5	7.5	x	30	15	y	15	x	15	4.5	7.5	4.5	x	15	4.5	3	x	4.5	4.5	22.5	x			
U	Towers	7.5	7.5	10.5	30	15	3	15	1.5	15	4.5	7.5	4.5	1.5	15	4.5	3	x	4.5	7.5	22.5	x	3		
V	Blowdown and water disengaging drums	4.5	15	15	30	15	7.5	15	7.5	15	4.5	7.5	15	7.5	15	15	7.5	x	15	15	22.5	x	7.5	x	
W	Onsite [illegible]		4.5	4.5	7.5	4.5	3	4.5	1.5	4.5	4.5	x	4.5	1.5	6	3	1.5	x	4.5	4.5	4.5	x	1.5	4.5	x

2.2.5 EQUIPMENT LAYOUT/UNIT PLOT PLAN DRAWING – GUIDE LINES

The following are the guidelines generally followed while making an Equipment layout drawing.

a) Equipment layout/Unit Plot Plan shall be drawn in 1:50 or 1:100 scale.

b) A0 size drawing sheet should generally be used for equipment layout. If the area to be covered is small, A1 size can be used.

c) Place north arrow at the top right hand corner of the sheet to indicate plant north.

d) The area above title block to be kept free for general notes, legends, reference drawings etc.

   i) One of the general notes should establish how to ascertain the datum level.

   ii) The legends normally adopted are;

      a) TOS – Top of Steel
      b) TOC – Top of Concrete
      c) TOG – Top of Grout
      d) FGL – Finished Ground Level
      e) FFL – Finished Floor Level
      f) FS – Fixed Support
      g) SS – Sliding Support

   iii) Floor finish or floor treatment required should also be explained such as;

      a) AR Tiling
      b) Grating
      c) Chequered Plate

e) All equipments are marked with its equipment no. as appearing in equipment list & dimensions (diameter, height/length etc.)

f) All equipments centerline are located in plant building w.r.t. the column grid. For layout of outdoor plant / offsite

facility, the equipment shall be located by co-ordinates. Avoid duplication.

g) Conceptual layout, P & ID, vendor / fabricated equipment drawings are to be used as basic document for preparing equipment layout drawing.

h) Walkways, cutouts, pipe racks, floor drains, gutter, trenches, ramp etc. if applicable should be clearly marked in the drawing. Mark invert level in trenches preferred slope for ramp is 1:6.

i) For in house plant layout, the location of staircases, lift & other utility areas should be clearly shown.

j) In equipment layout sectional drawing, for each equipment its top most or bottom most elevations should be marked. Enough sections to cover each equipment shall be considered.

k) Orientation of equipment shall be clearly marked for all the equipments by orienting one of the major nozzles.

l) In case of reactors / tanks, the location of manhole / hand hole, SG/LG,LI etc. shall be at accessible position.

m) Equipment lifting cutout alongwith laydown area shall be marked clearly in the drawing.

n) Equipment planned to be installed in future shall be shown dotted.

o) For heat exchangers, tube removal / cleaning space shall be marked.

p) While locating the pumps care shall be taken to ensure that the NPSH requirement is met.

q) General notes are written on one of the drawings (first) and shall not be repeated on all layouts but reference shall be given.

r) Direction of north shall be maintained same for all the plans for the same plant / project.

s) If more than one drawing is required to cover a specified area, then the match line shall be indicated clearly with the reference drawings. Matching coordinates shall be clearly marked.

t) One of the general notes should specify the absolute level of the area covered with respect to the plot. The datum preferably should be considered as +100.00M.

u) The equipment load, operating or test load whichever is maximum shall be considered for design and the layout should indicate this along with the dynamic factor wherever applicable. This could also be covered in table as well.

v) For reactors with agitators, lifting beam shall be provided for agitator removal.

w) For vendor equipments maintenance space as recommended by them for maintenance shall be provided.

x) Equipment layout shall also indicate the positions of utility stations, safety shower and eye wash.

y) Equipment elevation shall be so arranged to ensure gravity flow where specified.

2.2.6 TYPICAL LAYOUTS

In terms of the equipment arrangement, the equipment layout (unit plot plan) can basically be divided into two configurations:

a) the grade mounted horizontal arrangement as seen in the refineries and petrochemical plants, and

b) the vertical arrangement found in many chemical process industries.

Irrespective of the type of arrangement, there are certain basic principles to be followed while locating the equipment.

i) Economic piping: To minimize cost of piping; equipment should be located in process sequence and close enough to suit safety needs, access requirements and flexibility. Identify the group of

SG - Sight glass
LG - Light Glass

equipment that form a subsystem within the unit. The component within the subsystem to be arranged to have the most economical piping and the whole subsystem to be placed within the unit to have most economic inter connection.

ii) Process requirements: The equipment layout should support requirement such as minimum pressure drop, gravity feed and loop. The Piping Engineer should discuss such requirements with the Process Engineer before proceeding with the arrangement.

iii) Common Operation: The equipment that shares common maintenance facilities, common utility and continuous operator attention should be located in the same area. Although this may turn out to be expensive in terms of piping arrangement; the use of common building and equipment handling facilities will make up for the difference in cost.

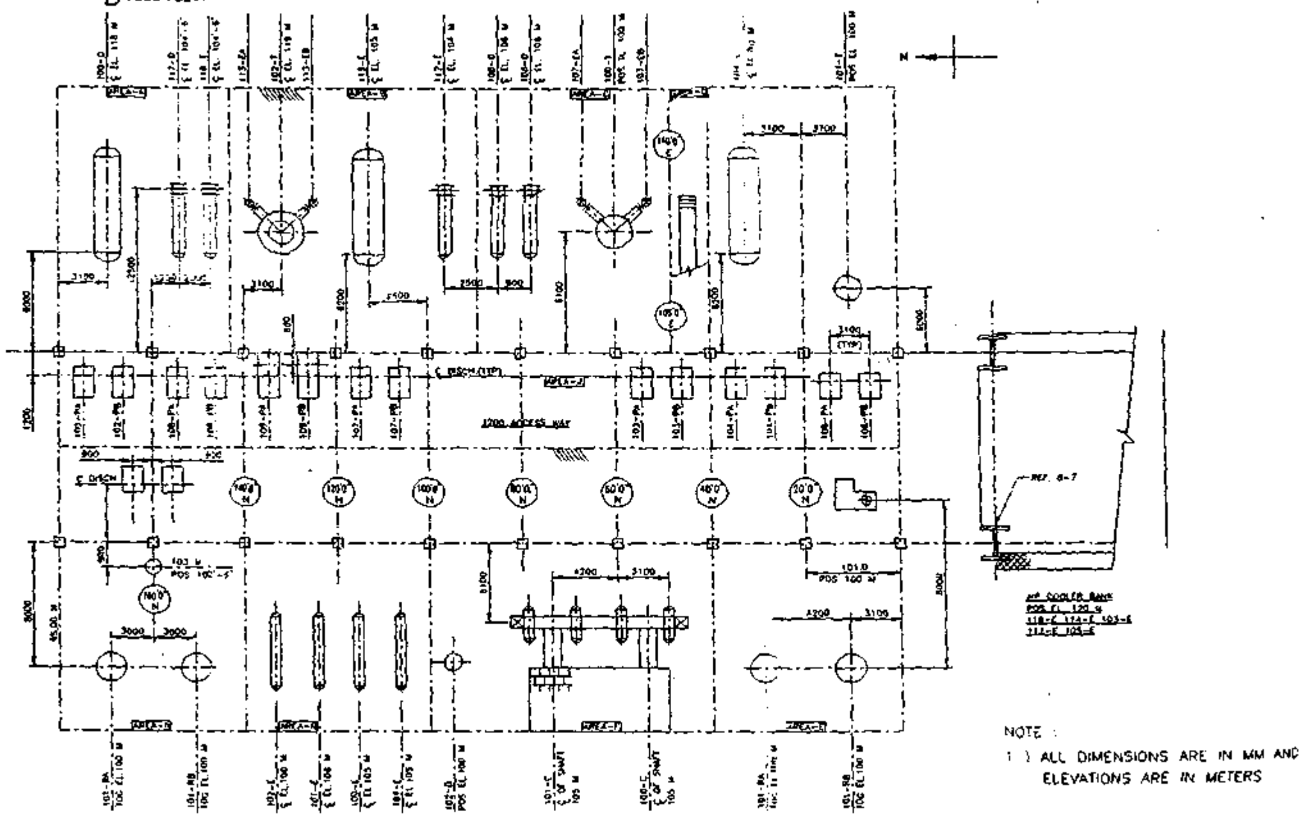


Fig. 2.2.1 TYPICAL PLOTPLAN OF AN OUTDOOR PETROCHEMICAL PLANT

iv) Underground facilities: Piping Engineer should investigate the facilities such as storm water drain, effluent drain, fire water, cooling water to be placed underground before deciding the equipment position. Depending upon the soil condition, the foundation will be either piled or spread footings. Spread footing foundation will require more space and equipment should be spaced to suit. In certain cases multiple equipment could be placed on a common foundation.

v) Climatic conditions: Weather condition influences the type of enclosures and location of equipment.

Wind influence the location of furnaces, cooling towers and stacks

a) Grade mounted Horizontal arrangement

In the grade-mounted horizontal in the line unit, the equipments are placed on either side of the central pipe rack with auxiliary roads. The main advantage of this arrangement is that the equipments are located at grade level, which makes it easier to construct, operate and maintain. The disadvantage is that it takes a lot of ground area. The typical layout of a grade mounted outdoor petrochemical plant is illustrated in Fig. 2.2.1.

The cross section of such units is shown in the sketches 2.2.2 to 2.2.5. In principle all these variations of the layout shown therein are same. One or two lines of process equipments are placed along the pipe rack. Maintenance roads are provided parallel to the pipe rack and process equipments. Central pipe rack is economical as shown in Fig. 2.2.2 and 2.2.3. Air coolers can be placed over the pipe rack while those at the ground will increase the ground coverage. When pumps are lined up under the pipe rack with central access and air coolers placed above, the insurance requirement may ask for sprinklers above the pumps. This type of layout is the most economical and thousands of petrochemical plants are built all over the world using this principle.

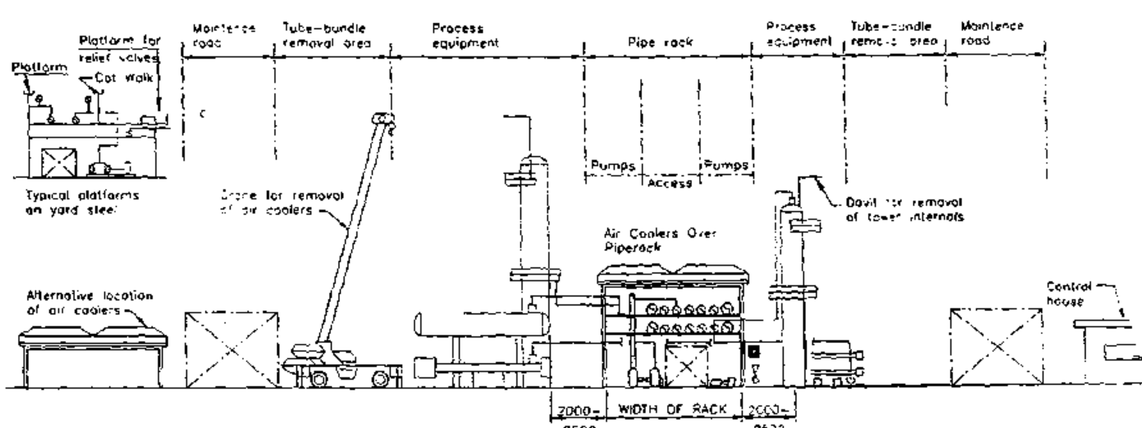


Fig. 2.2.2
TYPICAL CROSS SECTION OF AN OUTDOOR PROCESS PLANT

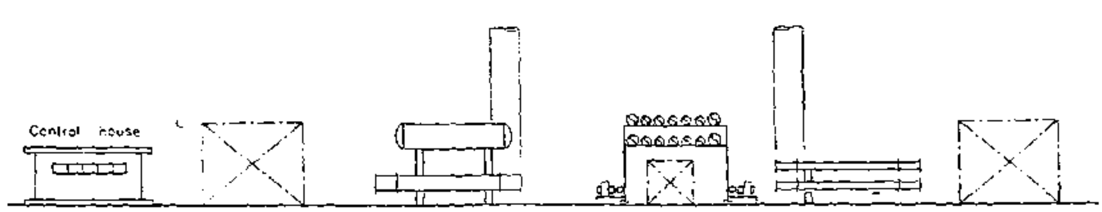


Fig. 2.2.3

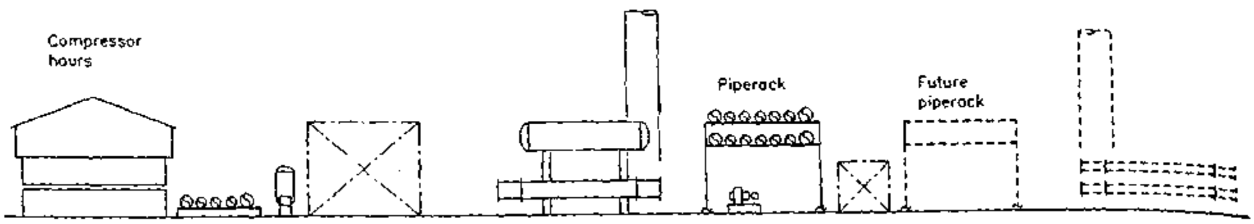


Fig. 2.2.4

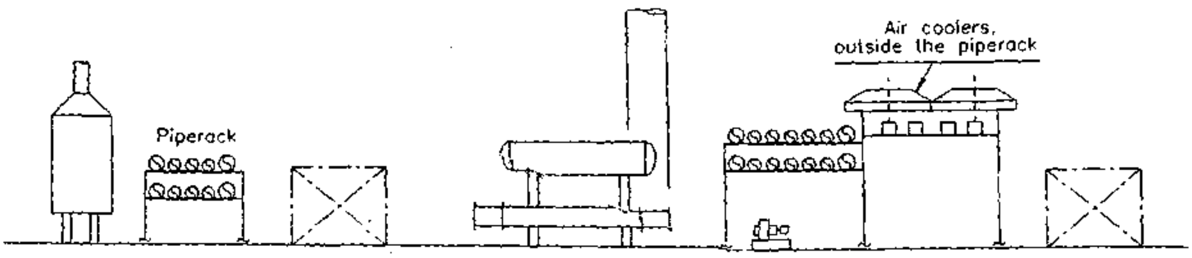


Fig. 2.2.5

When pumps are placed outside the rack that will increase the distance between the pipe rack and process equipment resulting in additional pipe length. This arrangement is shown in Fig. 2.2.3.

The one sided arrangement as shown in Fig. 2.2.4 and 2.2.5 are more expensive, since only one side of the area is used to locate the process equipments. However, if only a narrow area is available and or if expansion is to be taken up in future, these arrangements give optimum solution.

The control room and furnaces are placed outside the main process area, keeping the required safe distances. Auxiliary pipe rack is required to run cables to control room and piping from furnace to process area. Safety distance and maximum allowable length of transfer line influence the furnace location.

The basic principle to be remembered while locating equipments in all these cases is to eliminate, combine and minimize structures to achieve cost savings.

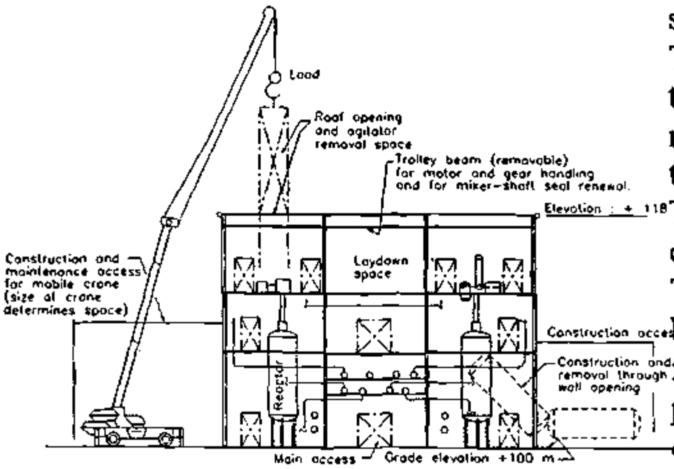
b) <u>Structure mounted Vertical Arrangement</u>

The structure mounted vertical arrangement has equipment located at multilevels in steel or concrete structure. This could be indoor or outdoor. The advantages of this are small amount of ground coverage and the ability to house the facility to suit process requirements or climate conditions. The basic principle of locating equipment in an indoor and outdoor plant remains same. The applicable principle is economy. In a multilevel layout, the vertical relationship of equipment also to be considered. The confined building does not change the philosophy of equipment layout and piping design. However, the requirements of operation and maintenance differ.

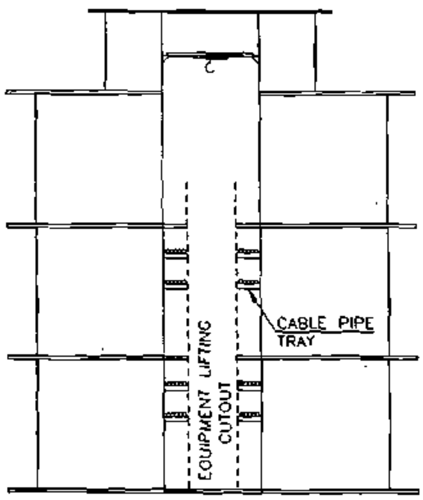
In a building, mobile platforms can be used extensively but it is not practicable outdoors. Hence permanent local platforms are more common in outdoor plants. Mobile cranes can be used in outdoor for maintenance. In indoor, tube pulling area for exchangers is to be built in. Building will be costly if this facility is to be included. Or else, the exchanger will

have to be shifted to workshop for cleaning which again will need more time and hence cost. Removal of large vessels, glass lined kettles etc., will need space above or below and also access aisle to outside with adequate clearance. Lifting beams with cranes located with negligible initial cost can make substantial savings in future maintenance cost.

Typical in house vertical arrangement is shown in Fig. 2.2.6.



TYPICAL CROSS SECTION OF AN INDOOR PROCESS PLANT
Fig. 2.2.6a



TYPICAL CROSS SECTION OF INDOOR CHEMICAL PLANT
Fig. 2.2.6b

2.3 Piping Layout

2.3.1 PHILOSOPHY OF YARD PIPING

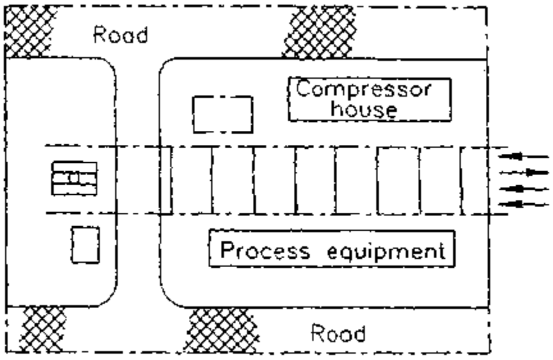
The main artery of an outdoor process plant is the pipe rack. Because the rack is located in the mid of an outdoor plant, the pipe rack must be erected first. Hence the development of structural drawing becomes one of the early requirements in a plant. To pass on the data to the civil/structural group, a civil scope drawing showing the width, the column spacing and the design load is prepared. This load data should include, in addition to the dead weight specified per running meter, the thermal and occasional loading the piping will impart on to the structure. The latter will include forces and moments depending on the type of support provided. The structural designer reinforces such bays on the pipe rack to overcome these forces and moments. Hence, a proper planning is required in the initial stages of design itself.

So, the first step in the development of pipe rack is the generation of a line – routing diagram. A line – routing diagram is a schematic representation of all process and utility-piping systems drawn on a copy of plot plan or it could be planometric representation of the utility and process line diagrams. Although it disregards the exact locations, elevations or interferences, it locates the most congested area.

The pipe rack splits the plant area into convenient parts. The pipe rack takes various shapes such as 'straight', 'L', 'T', and 'C' or 'U'. This configuration is based on the overall arrangement and site conditions. Based on the incoming/outgoing lines and locations, the pipe rack is laid.

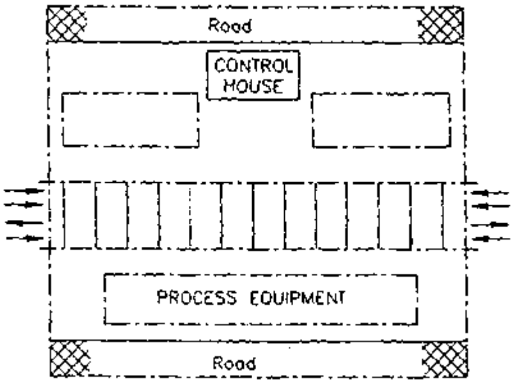
Fig. 2.3.1 to 2.3.7 shows the typical pipe rack layout for various plant arrangements. Smaller plants have the

simplest yard piping as shown in Fig.2.3.1 and 2.32. In the arrangement shown in Fig 2.3.1, the process and utility line enter and leave at one end of the battery limit. Fig. 2.3.2 presents a frequently adopted layout, with utility lines entering at one end of the battery limit and process lines at the opposite end. This is called a straight through yard. Layout condition sometimes result in an 'L' – shaped yard as shown in Fig. 2.3.3.



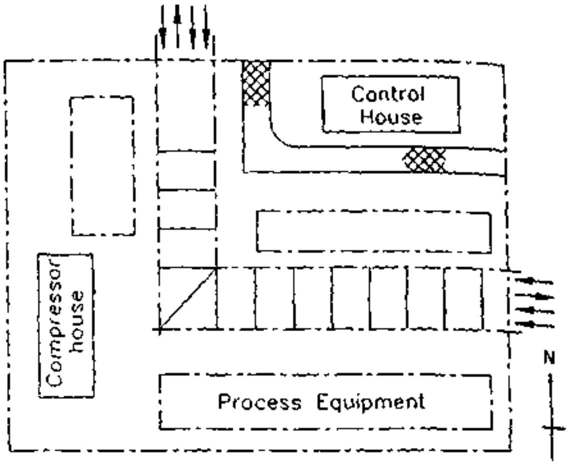
Dead–end yard. Lines enter and leave one end of yard.

Fig. 2.3.1



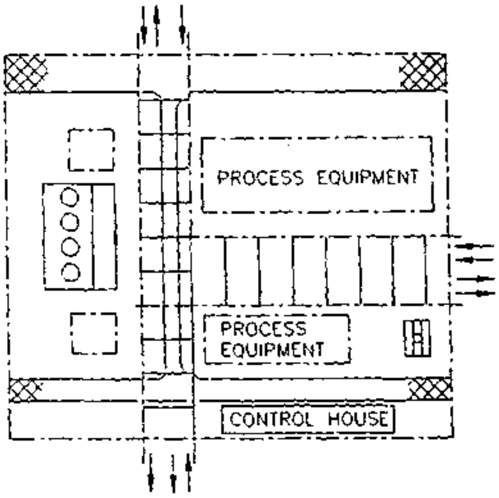
Straight – through yard. Lines can enter and leave both ends of the plot

Fig. 2.3.2



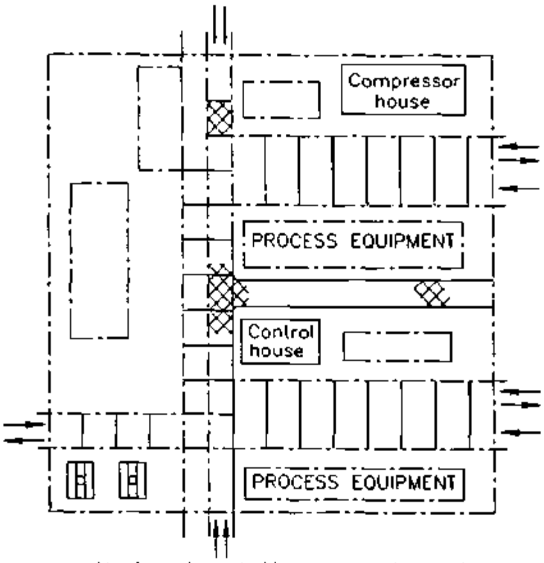
L–shaped yard. lines can enter and leave north and east side of the plot.

Fig. 2.3.3



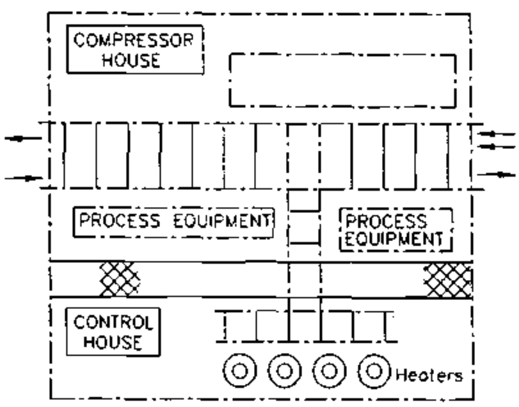
T–shaped yard. Lines can enter and leave on three sides of the plot

Fig. 2.3.4



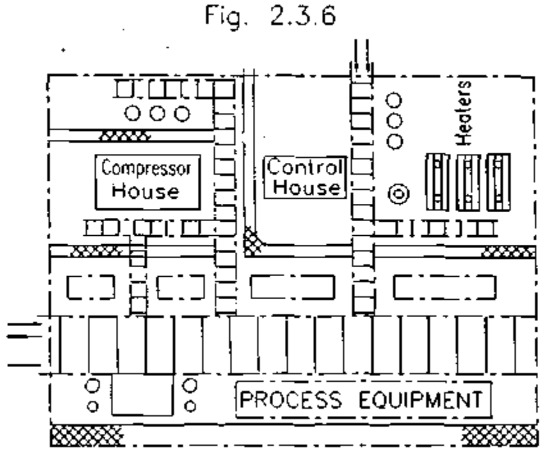
U-shaped yard. Lines can enter and leave all four sides of the plot

Fig. 2.3.5



Combination of L- and T-sahped yard..

Fig. 2.3.6



Complex yard-piping arrangement for a very large chemical plant.

Fig. 2.3.7

In larger plants, yard piping will be more complicated as shown in Fig. 2.3.4, 2.3.5 and 2.3.6. Fig. 2.3.4 shows a 'T' – shaped yard. Process and utility lines can enter and leave on three sides of the plant. Fig. 2.3.5 shows a 'U' or 'C' – shaped yard. Lines can enter and leave at all four sides of the plant. Fig.2.3.6 shows a combination of the 'L' and 'T' – shaped yard. Lines enter and leave similar to that shown in Fig. 2.3.2. Fig. 2.3.7 shows complex yard piping for a very large plant. This layout can be considered as a combination of many simpler yard piping arrangements.

Of course, the configuration of pipe rack is not determined while doing the plant layout. The arrangement results from an overall plant layout, site conditions, client requirements and above all plant economy.

The pipelines on the rack are classified as process lines, relief line headers and utility lines. The rack should accommodate the electrical and instrument cable trays as well. The width of the pipe rack is estimated as

W = (f x n x s) + A + B

Where

f = Safety factor

= 1.5 if pipes are counted from the PFD

= 1.2 if pipes are counted from P & ID.

n = Number of lines in the densest area up to the size of 450NB

s = 300mm (estimated average spacing)

=225mm(if lines are smaller than 250NB)

A = Additional width for

(1) Lines larger than 450 NB

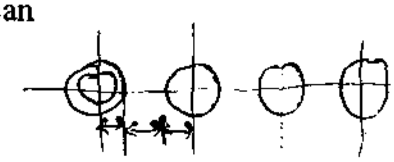
(2) For instrument cable tray/duct

(3) For electrical cable tray

B = Future provision

= 20% of (f x n x s) + A

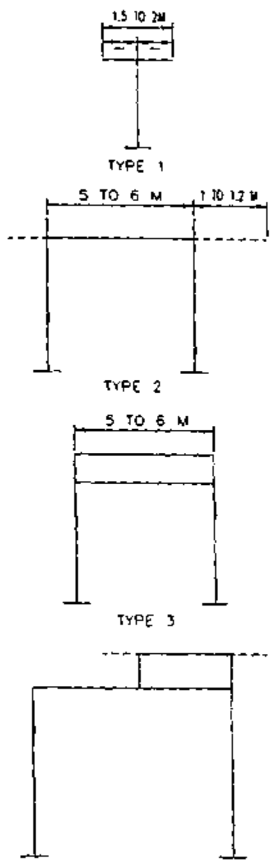
Normally pipe rack width is limited to 6M. If the width worked out thus is more, then the arrangement to be done in multiple layers. The space requirements of equipment along with the access below influences the width of the rack. The arrangements adopted are:



½ dia. of bigges flange + ½ dia. of other pipe + insulation thickness

a) Single column rack 'T' type
b) Double column rack with a single tier
c) Double column rack with a double tier

These are illustrated in Fig. 2.3.8.



TYPES OF RACKS
Fig. 2.3.8

Depending upon the type of plant the rack could be of steel, concrete or a combination of both. The spacing between the bent/column of the pipe rack is normally 5 to 6 meters. Wide spacing is necessary at road crossings or where loading and access space are needed. The headroom clearance also depends upon the type of crossings.

min height of Rack is 3 m.

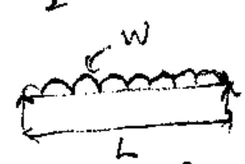
The headroom normally provided is as below:

Sr. No.	Description	Head room (mm)
1.	Clear headroom under Structures/pipe lines inside operating area.	2200
2.	Head room over rail (from top of rails)	7000
3.	Clear headroom above crest of road for crane movement.	7000
4.	Clear headroom above crest of road for truck movement.	6000
5.	Clear headroom above crest of road between process units.	4500

A typical arrangement of yard piping is illustrated in Fig. 2.3.9.

The yard piping can also run on the sleepers. These are made of concrete and are mainly used to run large diameter piping such as cooling water piping. Another way to run the yard piping is in open trenches with the arrangement same as that of a single tier rack. The road crossings are done by culverts. Water logging in the trenches and draining of the same are major problems to be handled in this case.

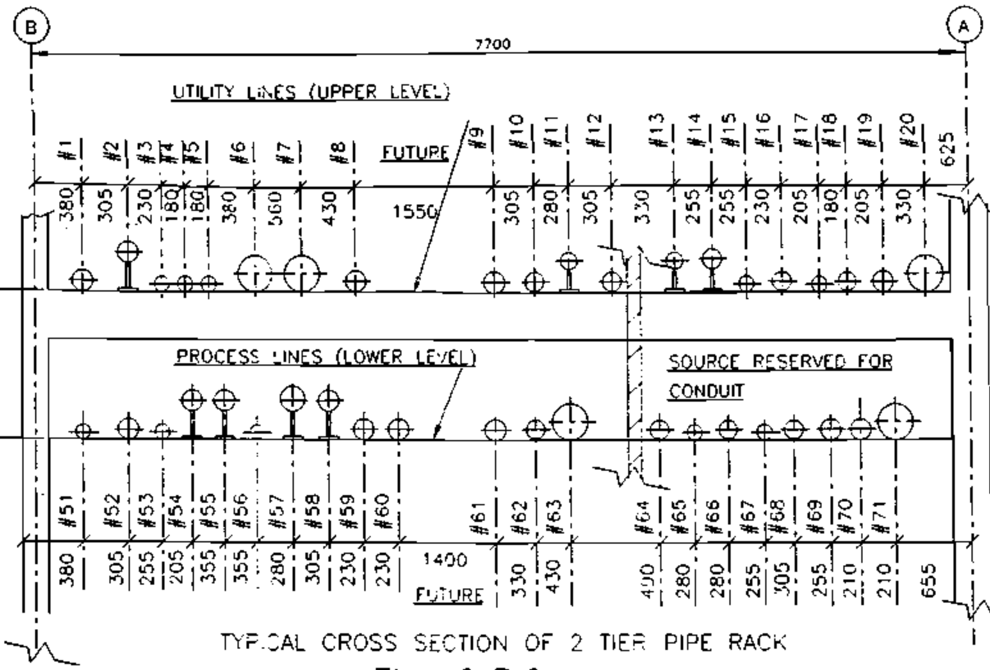
$$\frac{M}{I} = \frac{f}{y}$$



$$M_{max} = \frac{Wl^2}{10}$$

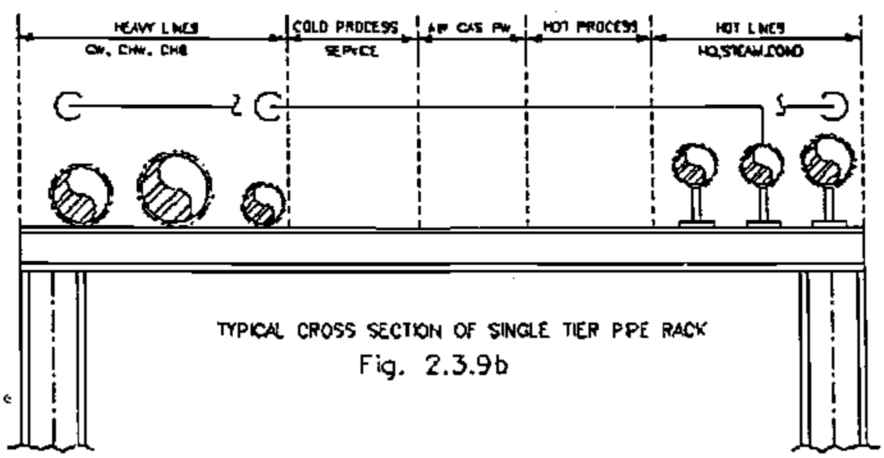
$$M = fZ$$

$$f = \frac{Wl^2}{10}$$



TYPICAL CROSS SECTION OF 2 TIER PIPE RACK

Fig. 2.3.9a



TYPICAL CROSS SECTION OF SINGLE TIER PIPE RACK

Fig. 2.3.9b

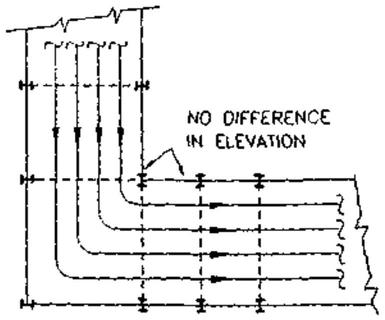
2.3.2 PIPING ARRANGEMNT

P & I diagram, equipment layout, piping specifications, equipment drawing and the vendor requirement for proprietary equipment form the basis of a piping layout. In areas where piping is critical, the equipment locations are fixed only after a 'piping study' is made. This will facilitate access to equipment after piping is in place and also to have the most economical pipe routing.

Fig. 2.3.9 shows typical cross section of pipe rack for a petrochemical plant. Piping shall be arranged in an orderly manner and routed as directly as is practical in established pipe racks. As far as practical, piping should run at different elevations along north-south and east-west directions. The basic principle to be followed in such cases is **change in direction, change in elevation.** Combined change in direction and change in elevation is effected by 90^0 elbows. To achieve minimum change in elevation, combination of 90^0 and 45^0 elbows may be used. (see Fig.2.3.10 and 2.3.11)

In cryogenic gases & liquids, tapping is done from the top.
In liquids. It is from bottom

 every 30 to 40 mt, there is a loop



TURN IN PIPERACKS
(CHANGE IN ELEVATION WHEN
CHANGING DIRECTION)
Fig. 2.3.10

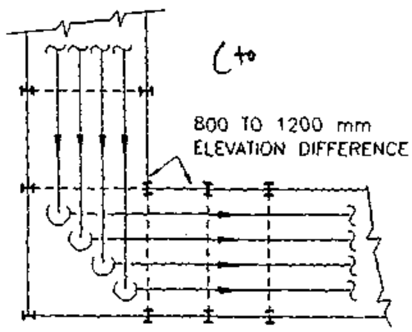
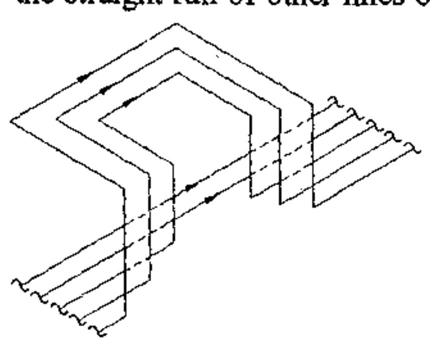


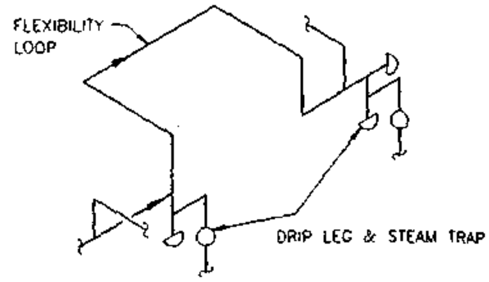
Fig. 2.3.11

Although the final stress analysis will be done after the routing are finalized, preliminary check from monogram is required to establish the requirement and size of expansion loops. The flexibility temperature can be obtained from the line list. The lines requiring expansion loops is placed on the top level of the pipe rack. The line that requires the largest expansion loop leg must be located on the outside. The loop shall be arranged with the portion of the loop getting elevated from the rack level by two 90^0 bends. This will allow the straight run of other lines on the rack and also provide additional flexibility (see Fig. 2.3.12)



GROUP OF LINES WITH EXPANSION LOOPS
(HOTTEST AND LARGEST LINE OUTSIDE)
Fig. 2.3.12

Proper drip leg and steam traps shall be provided for steam lines to avoid the condensate collection at these points (see Fig. 2.3.13).



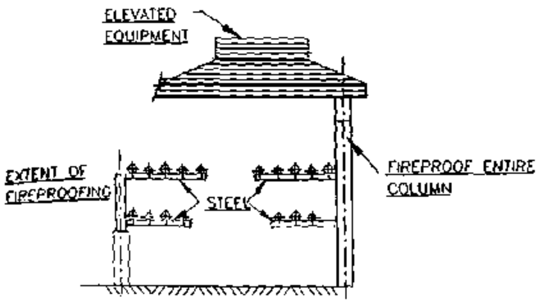
DRIP LEG & STEAM TRAP
Fig. 2.3.13

Header growth can create problem of interference, which is often overlooked. The spacing on the rack should account for these. The line, which expands also, should be properly guided to ensure maximum effectiveness of expansion loops.

Most of the lines leaving or entering the rack perpendicular to it need support. This is affected by the provision of the structural members, known as spandrels, connecting between the columns at the required elevation. These spandrels could be located along the external face of the column or on the centerline of the same.

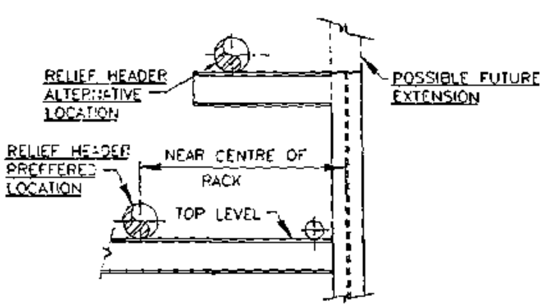
If hydrocarbons are prevalent in the plant, it is a common practice to fireproof the columns just below the lower support rack support beam. If air coolers or any other equipment is located above the pipe rack, the fireproofing is extended up to the equipment support beams (see Fig. 2.3.14). Fireproofing is done on the rack columns by covering these by Plain Cement Concrete (PCC).

on Racks. generally utility piping is on 1st rack, process on 2nd rack. And electrical cables on top.



FIRE PROOFING REQUIREMENTS OF RACK COLUMN
Fig. 2.3.14

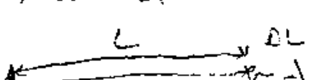
The expansion of the rack should also be thought while planning. It is normal practice to add additional tier on the top. To accomplish the same, the space above the column should be kept free of piping or conduit. The preferred location of relief header is as shown in Fig. 2.3.15.



LOCATION RELIEF HEADER
Fig. 2.3.15

Alternative to the expansion above the rack could be planned for expansion outside which not preferred. This keeps the process equipments away. (Refer Fig. 2.3.16)

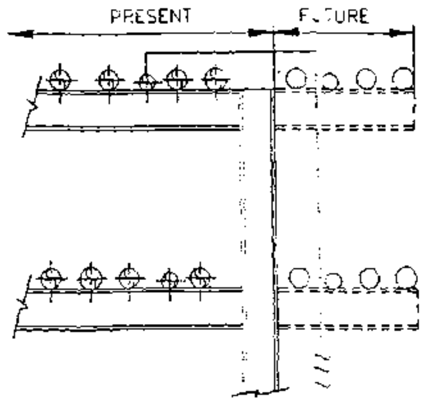
every 60 mt Rack, expansion Joints are Provided.



$\epsilon = \Delta L / L$

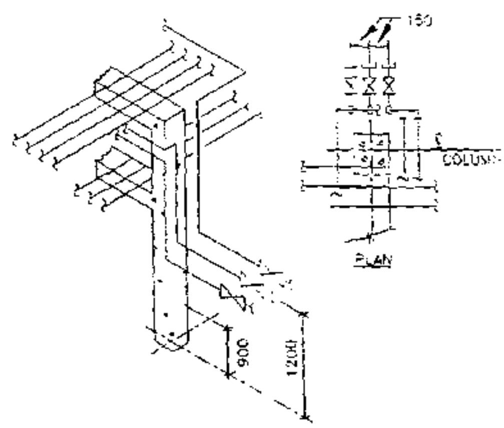
$f = E\epsilon$

$P = AF$



EXTENSION OF PIPERACK
Fig. 2.3.16

The typical arrangement of hose station along the rack is as shown in Fig. 2.3.17.

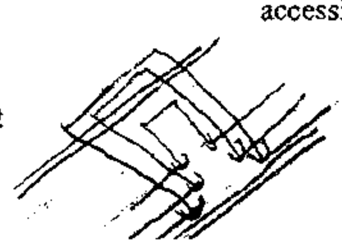


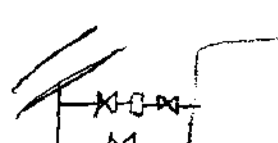
UTILITY STATION ALONG RACK
Fig. 2.3.17

The overall pipe rack design must meet the current needs as well as expansion plans without major modifications. Heavy large diameter piping may be laid below ground or on sleepers as established by economics.

2.3.3 VALVES-LOCATION

Accessibility to valves and instruments should be the primary concern while arranging the same on the piping. Process isolation valves should be readily accessible, the valve-stem centerline being





at an elevation of 1200 - 1500 mm from the operating level. If at elevated level, they can have chain operators. Valves located at low or high locations can have extended stems to reach the access aisles. One thing must be ensured that the **stem will not be oriented below horizontal level. (Fig. 2.3.18)**

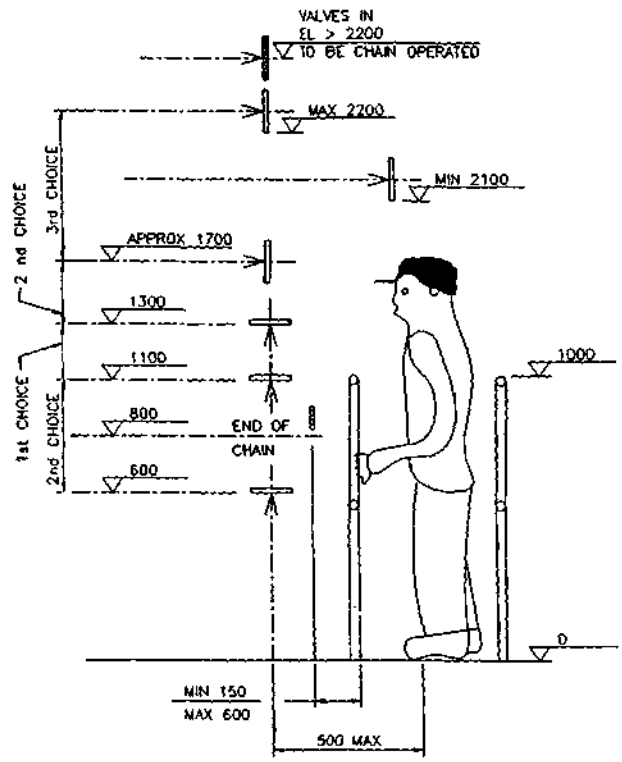


Fig. 2.3.18

Utility lines have two groups of valves. One group is closely related to process and should be located having accessibility just like process valves. The other groups contain all isolating valves such as block valves on headers, root valves on sub headers and valves for future connections. These valves are seldom used and are located overhead. Temporary ladder accessibility is acceptable for such valves (see Fig. 2.3.19). Battery limit isolation valves can be provided with access platforms, valves being located on either side of the platform to economize spacing. Battery limit valving for a single tier pipe rack is shown in Fig. 2.3.20. The valves are staggered on either side of the catwalk. When a pipe rack enters a unit, the elevation changes may be required. The block valves could be installed on vertical leg in this case as shown in Fig. 2.3.21. This allows relatively easy operation Fig. 2.3.22 shows two level racks with elevation charges above or below rack level. High point vents and low point drain valves are provided to avoid frustration during testing. These shall be piped to accessible spots.

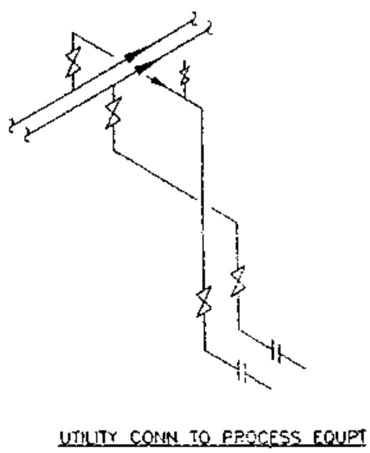


Fig 2.3.19

In case of Flare header, the valve is kept with handwheel down.

Instead of elbow Tee shd be provided to collect condensate

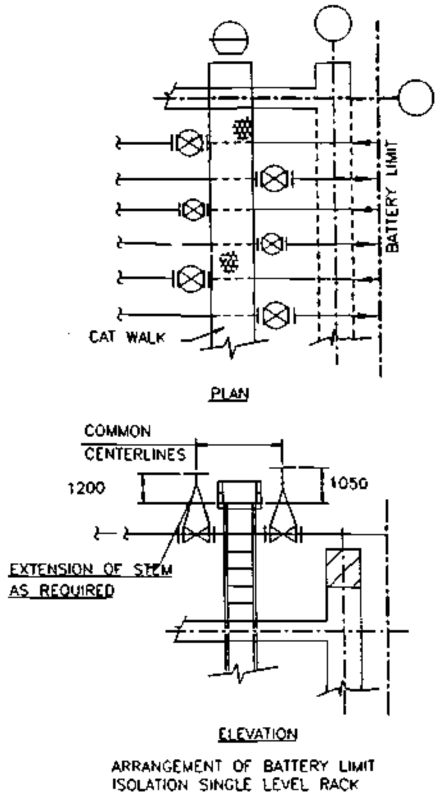


Fig 2.3.20

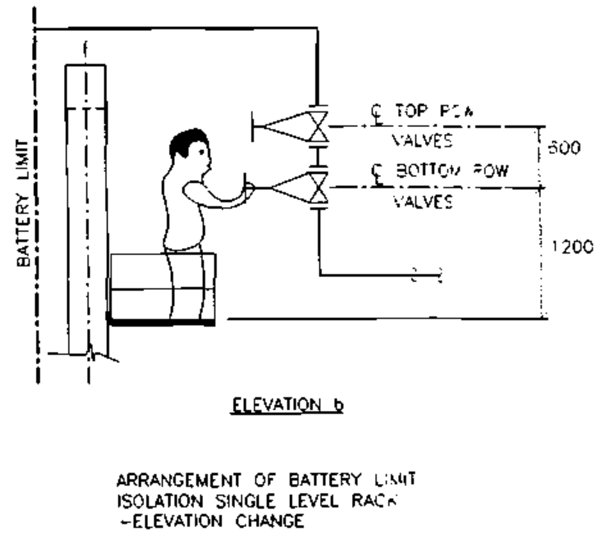


Fig 2.3.21b

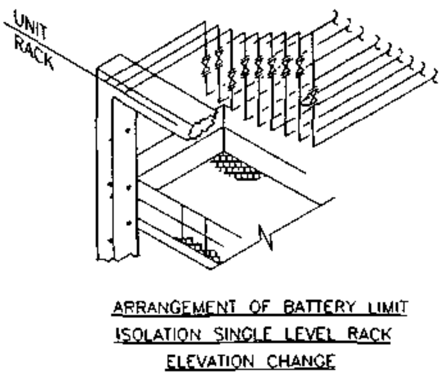


Fig 2.3.21a

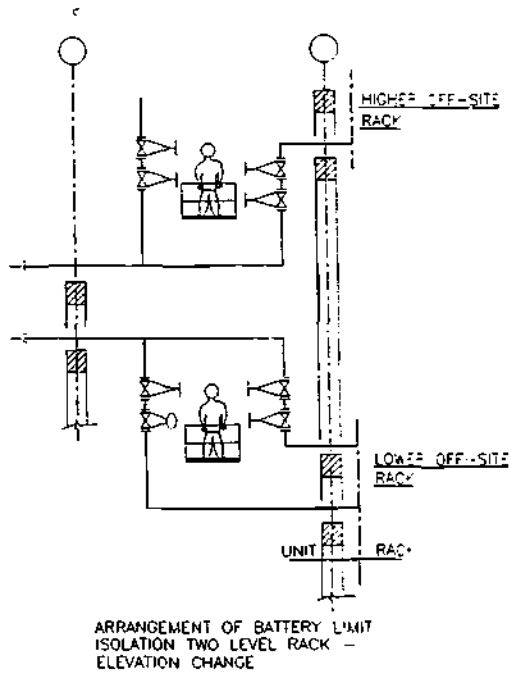


Fig 2.3.22

2.3.4 ELECTRICAL/INSTRUMENT TRAYS

The Electrical and Instrument cable tray/duct location is coordinated in the design-planning phase itself and integrated in the overall arrangement.

Space for these trays is provided above the piping on the pipe rack. Future

provision for these is also taken care of while designing the pipe rack. Where the requirement of pipe rack does not exist, cable rack is provided separately as required. Specific locations are allotted, including the type of support so that electric and instrument hardware and their support do not interfere with the operation and maintenance access.

2.3.5 PIPING FOR INSTRUMENTS

Instruments,when located on piping ,will need certain specific requirement for it to perform the duty for which it is provided. Piping Engineer should be aware of these requirements and should take care of the same while routing these pipe lines.

a) Flow measurement instrument need certain straight length upstream and downstream of the instrument.This is normally 15D on the upstream and 5D on the downstream.

b) The pipe lines in which flow meters such as magnetic flowmeters ,vortex meters ,turbinemeters etc are located should be routed in such a way that the line will be full with liquid all the time.The pipe line should be supported on both sides of meter.

c) Control valves are located at grade, at about 500mm height to provide convenient access for operation and maintenance. Block and bypass valve also form the same criteria. The standard arrangements followed are as per Fig 2.3.23. If pocketing the process line is unacceptable, then a permanent or mobile platform should be planned, as access is very important. Locating control values on the vertical line should be avoided.If is unavoidable; the should actuator should be supported properly.The bypass should be selected for easy operation.

d) Isolation valves for level gauges and pressure gauges shall be made accessible. Access and space for the removal of level controllers temperature probes ,conductivity probes,bottom flanges of the control values etc shall be provided. All primary and secondary indicators of pressure, temperature, flow, level, positioners etc. should be visible from the operating area.

e) Rotameter shall be placed on vertical line and the inlet should be from the bottom of the instrument.

f) Thermowell shall be located on the pipe line of required size. Instrument hook up shall be referred for the requirement.

g) Enough operating and maintenance access shall be considered while locating any instrument.

2.3.6 SAFETY VALVE PIPING

Safety valve is defined as automatic pressure retrieving device actuated by the static pressure upstream of the valve and characterized by full opening pop action. It is used for gas or vapour service.

Relief valve is defined as an automatic pressure relieving device actuated by the static pressure upstream of the valve which opens further with the increase in pressure over the opening pressure. It is used primarily for liquid service.

Safety Relief Valve is an automatic pressure actuated relieving device suitable for use either as Safety Valve or Relief Valve depending on application.

a) SRV shall be installed in upright position and directly attached to piping system or equipment to be protected.

b) Depending on the service the system can be open discharge or closed discharge.

c) The installation should be such as to keep the piping force to the minimum. To reduce the force/stress in the inlet line;

i) The inlet line should be kept more than the valve inlet size.

ii) The inlet nozzle can be reinforced.

d) The inlet line for liquid service shall never run dry. Formation of bubble at inlet shall be avoided.
e) Supporting arrangement shall be such that the piping should not vibrate while the valve is discharging.
f) Gas/Vapour discharge line should be kept atleast 3M above the nearby platform.
g) Liquid discharge line shall be routed to facilitate draining the system.
h) A weep hole shall be provided for the vapour discharge line.

2.3.7 RUPTURE DISC PIPING

a) Rupture Disc shall be installed at the direct vicinity of the system to be protected.

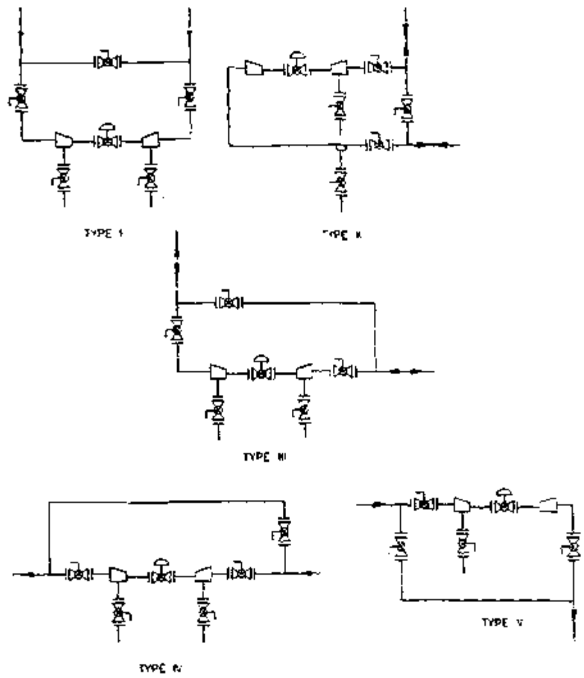
b) Discharge piping should be kept as short as possible to ensure safe discharge of the fluid.

c) If rupture disc is of graphite, trap shall be installed downstream to collect the broken pieces.

d) It will be ideal to integrate the rupture disc into vertical lines with flow from the bottom.

2.3.8 MISCELLANEOUS

Space requirement of HVAC ducting, where applicable, shall be integrated into the layout at the stage of development. The equipment for these will be housed separately and should include in the overall plot plan. The dimensional requirements of ladders, stairways, trenches, platforms etc. are to be considered while making equipment arrangement and planning access to valves and instruments.

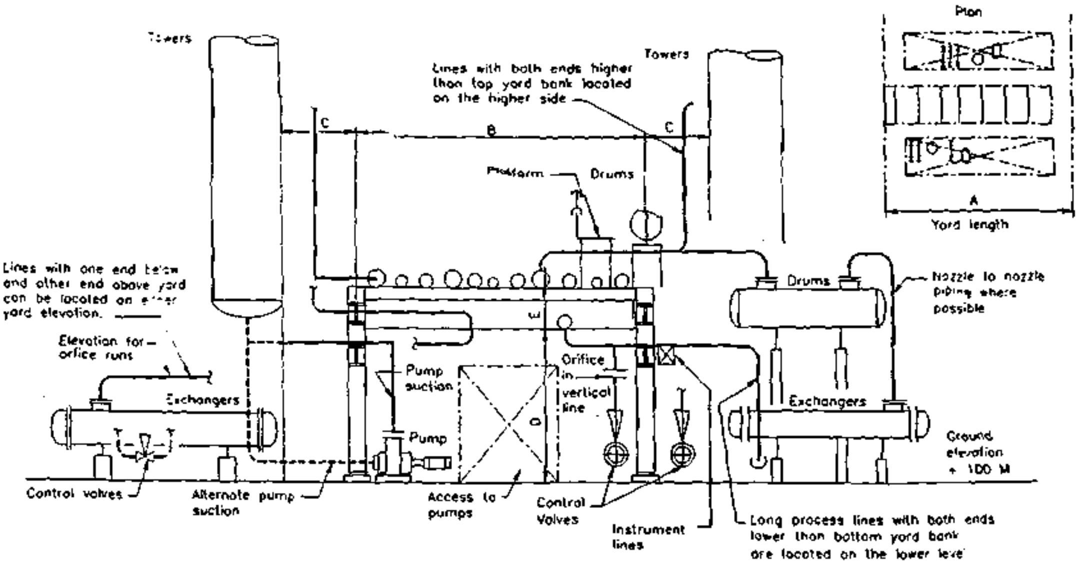


CONTROL VALVE ARRANGEMENT
FIG. 2.3.23

Figure 2.3.24 shows cross section of a typical yard piping for petrochemical plant. The various structural elevations are determined by the guidelines for the requirement of headroom given above. Size of the supporting beams should also be taken into account while finalizing the structural levels.

Generally, process lines that connect two lines located at a level higher than the top tier should run on the top tier of the rack. Lines having one end located at elevation lower than the bottom tier can run in the top or the bottom tier. If both ends of a line are lower than the bottom tier, the line should run on the bottom tier of the rack.

The elevation of lines can be influenced by valve and instrument location. Access platforms are required to be provided if valves are to be placed on the top at rack level. The preferred location of lines with orifice runs is near



TYPICAL PIPE RACK CROSS SECTION FOR PIPING ARRANGEMENT

Fig. 2.3.24

the edge of the yard, with orifice flanges near a rack column for access, with a portable ladder. Orifice runs after pumps can be located near supporting column at 2.5 m level. Control valves are usually located near rack columns for convenient support.

Keeping dimensions B and C to the lowest required levels will minimize the length of pipe between rack and process equipment and for connecting equipment on opposite sides of rack. The dimensions D and E at not more than the required yard height will reduce the vertical pipe runs. However, the distance E is kept 1 to 1.5 m to have a proper erection and maintenance access and depends on the size of the pipe at the lower tier.

2.3.9 STATUTORY REQUIREMENTS

The layout designer should familiarize himself with the law of the land while planning the equipment and piping arrangement. The requirement as per the following shall be adhered to:

a) The Factories Act 1948.
b) The Petroleum Act 1934 & The Petroleum rules 1976.
c) The Static and Mobile Pressure Vessels (unfired) Rules 1981.
d) The Gas Cylinders Rules 1981.
e) The Indian Boiler Regulations 1951.
f) Development control rules by the State Industrial Development Corporation.

2.3.10 CRITICAL EXAMINATION TECHINIQUE

The quality of the equipment and piping layout can be established by the Critical Examination Technique where you ensure that all the following parameters are well addressed

a) It is process adequate?
b) It is operator friendly?
c)It is construction clear?
d)Has adequate maintenance access provided?
e)How to evacuate in case of emergency?
f)Has safe fire fighting access provided?
g)Standard practices where applicable has been adopted?
h)Is the piping arrangement aesthetic ?
i)Is supporting arrangement adequate and aesthetic ?
j) Is piping adequately flexible ?

3.0TYPICAL ANALYSIS

With the general principles and requirements as described above, we will analyze the layout and piping design of specific equipment deployed in chemical/petrochemical process plants.

3.1 Pumps

Pumps rarely influence the plant layout except where a common standby for two services or multiple duty pumps might dictate the process equipment arrangement. But the pumps can never be treated as an independent entity, but to be treated as part of the piping system, which affects the performance even if the basic selection is faultless.

Types of pumps

Before looking into the layout and piping details of pumps, we will review the pertinent details of the equipment that affect the plant layout and piping design. Basically there are two types of pumps, the centrifugal type and the positive displacement type.

The centrifugal type could be a single stage or a multistage. The single stage pump has one impeller and multistage pump has two or more impellers in series. The discharge of one impeller is the suction of the next one and the head developed in all the stages are totaled.

Based on the suction and discharge arrangement the type of pumps available are

(a) end suction top discharge
(b) top suction top discharge
(c) side suction side discharge.

The end suction top discharge pump has vertically split casing and the end suction end discharge pump has horizontally split casing. Vertically split casing has good maintenance access. They are normally the back pull out design, which facilitates the removal of impeller without affecting the piping connections. Pumps will be mounted on a base plate with the motor, keeping the motor shaft and pump shaft carefully aligned. This base plate will be grouted on to a concrete foundation. The size of this foundation will be approximately 500 x 1500 mm or can vary up to 2000 mm long in case of large pumps.

Large capacity water pumps usually have horizontally split casing. Inlet and outlet are horizontal at right angles to the pump shaft. Suction piping should be short and straight with one or two expansion joints.

Inline pumps are compact and mounted along the pipeline even overhead. Large inline pumps may need separate supported on vertical shaft pumps occupy less area but needs head room for removal.

The positive displacement pumps can be rotary or reciprocating type. The rotary pumps work with forced volume displacement and can deliver constant pulsation free flow against higher head than the centrifugal pumps. The layout and piping design do not differ from that of centrifugal pumps.

Reciprocating pumps are used where very high head is needed for a low flow. Here the discharge cannot be throttled to obtain capacity control as in the case of centrifugal pumps. Instead variable speed drive or stroke adjustments is used. The alternating action of reciprocating pumps produces pulsation flow. The extent and frequency of pulsation depends on the number of cylinders in parallel and whether they are single or double acting. Due to pulsating operations, these pumps are bulky, but are considered for very low flow applications. The pumps could be single acting with single piston plunger with very high pulsating flow. The variation in cylinder arrangements gives

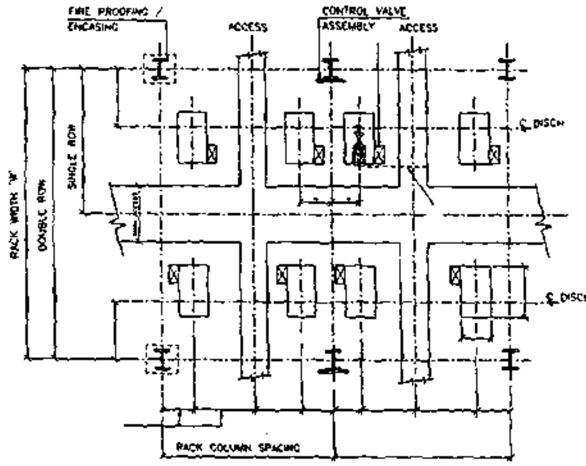
during commissioning
This type # temp.
SH lines used with spot

single, duplex, triplex and quadruplex pumps. The diaphragm pumps are compact and deliver precise quantities of fluid.

The positive displacements pumps are not suitable for pumping slurry and are not suitable for abrasive services.

3.1.1. EQUIPMENT LAYOUT

The design of equipment and piping configuration affect the energy used and capital cost of pumps. Hence, economy of piping and structures along with ease of operation and maintenance are the principal aim while arranging the pumps. Pumps are placed close to process vessels. Number of pumps should be lined up and aesthetically well arranged. Pumps are arranged under the pipe rack in a refinery or an outdoor process plant, keeping motor end towards the access space and suction/discharge faces towards the process vessels (Refer Fig. 3.1.1). Single pump should have access all around, a minimum of 900mm. When space is restricted, or the pumps are small, two pumps can be placed on common foundations with the orientation of the motor terminal modified if required. (Refer Fig. 3.1.2)



PLAN

LAYOUT OF PUMPS IN REFINERY / PETROCHEMICAL PLANT

Fig. 3.1.1a

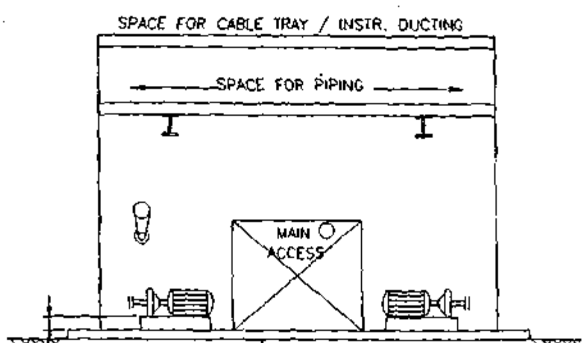
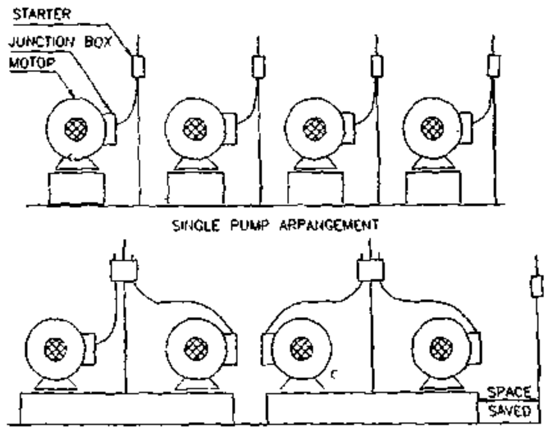
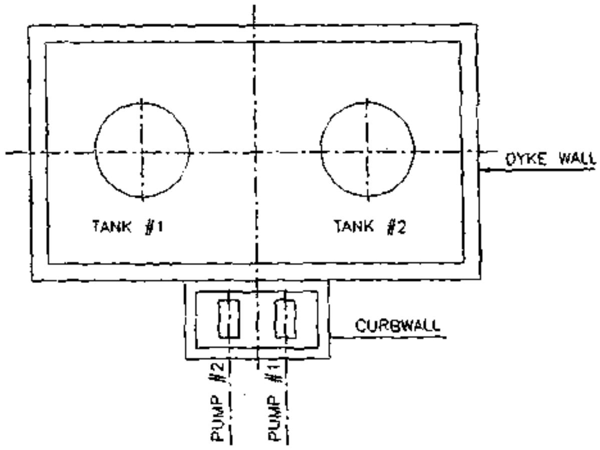


Fig. 3.1.1b



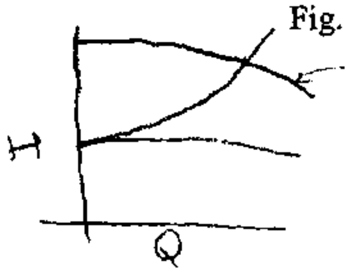
PUMP ARRANGEMENT

Fig. 3.1.2



ARRANGEMENT OF PUMPS IN A TANK FARM

Fig. 3.1.3



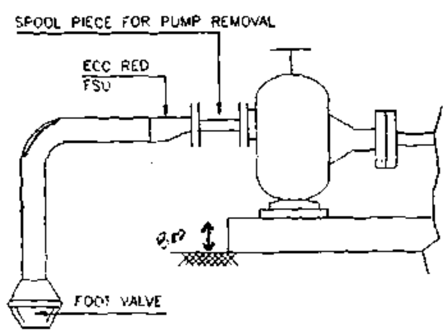
flange

tank farms

For safety and operators' convenience, pumps for tanks containing inflammable/corrosive liquids should be located outside the dykes. (Refer Fig.3.1.3)

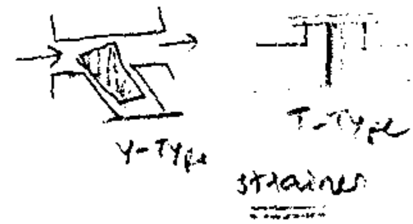
3.1.2 PIPING ARRANGEMENT

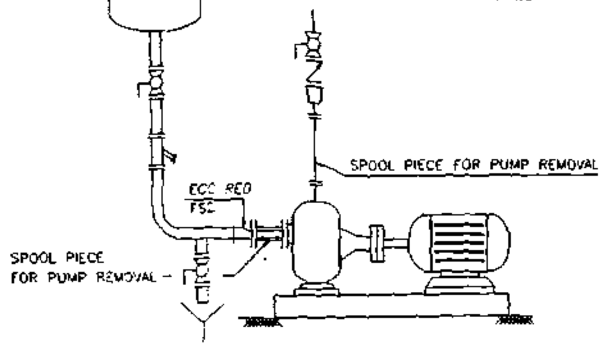
The primary goal in piping arrangement is to satisfy the performance and the flexibility requirements. Suction piping should be designed without loops or pockets. The eccentric reducers are placed close to the suction nozzle either FSU or FSD depending on the line configuration (Refer Fig. 3.1.4). These *lines shall be drainable near the pumps.* The suction line is generally one or two sizes larger than the pump suction nozzle for centrifugal pumps. NPSH requirement has to be checked while locating the pumps and routing the suction lines. Accordingly saturated liquid, steam condensate and vacuum conditions need elevated suction vessels. The horizontal runs should be kept minimum in all suction piping. The thermal expansion requirement should be taken care of in such a way that the pump nozzles are not loaded. The suction as well as the discharge piping shall be supported adequate enough not to impose excess forces and moments, due to dead weight and pipe expansion.



PUMP WITH SUCTION VESSEL BELOW

FIG. 3.1.4a

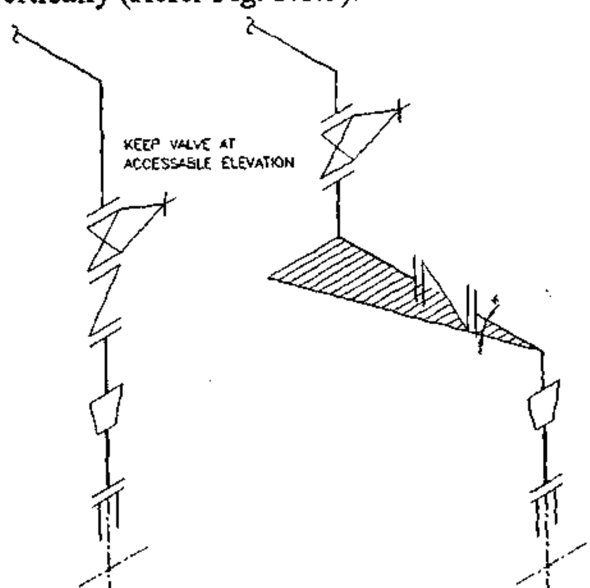




PUMP WITH SUCTION VESSEL ABOVE

FIG. 3.1.4b

The check valves are placed on the discharge piping to arrest the back flow and the reverse run of the impeller due to it. If the check valve is piston lift type, the piping should be arranged horizontally and if it is swing type it can be placed vertically (Refer Fig. 3.1.5).



DISCHARGE PIPING ARRANGEMENT

FIG. 3.1.5

Streamlined piping is desirable at suction and discharge piping of reciprocating pumps. Use long radius bends and angular branch connection to avoid sudden change

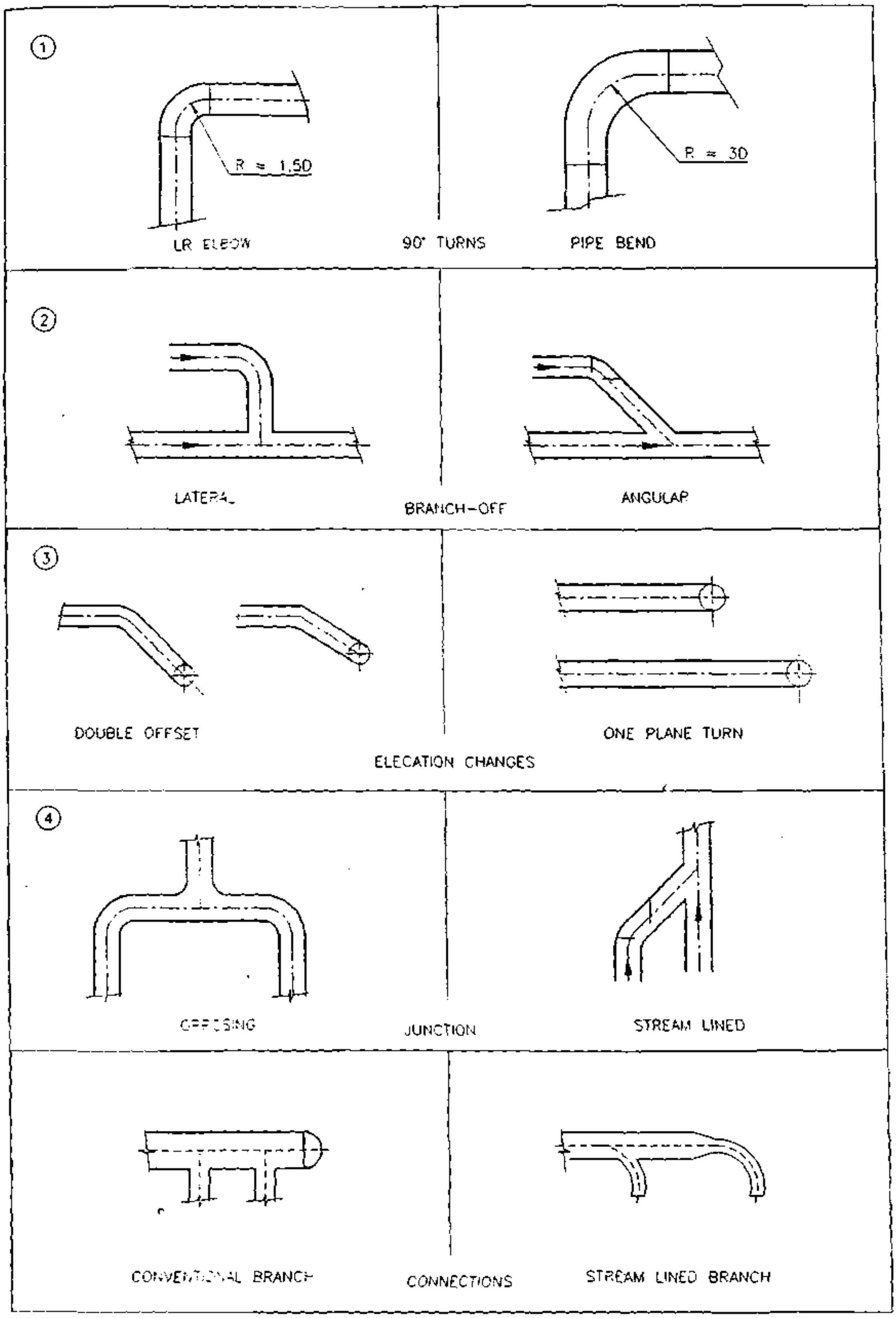
In case of pump, Centre line of discharge is located

of direction and opposing flows (Refer Fig. 3.1.6). Spool pieces shall be provided to facilitate pump removal because the maintenance requirement is relatively frequent. If the pumps are designed to do the multiple duty, the suction manifold and the discharge manifolds shall be arranged nearer to the pumps to make it operator friendly.

Auxillary manifolds to supply cooling water, seal flushing fluid, gland oil, for pump jackets heating/cooling shall be located overhead and nearer to the pump.

The complexity of piping system design, maintenance, and troubleshooting requires the Process Engineers, the Maintenance Engineers and the Piping Engineers on the same wavelength and work more closely together.

- In cooling water pumps, the expansion bellows are provided on inlet & outlet due to black body temp. of pump



PIPING FOR RECIPROCATING PUMPS AND COMPRESSORS

FIG. 3.1.6

3.2 Compressors

Basically there are two types of compressors used in the chemical process industry. They are the centrifugal type and the reciprocating type.

Layout and piping considerations for the centrifugal compressors do not differ in concepts from arrangement of centrifugal pumps. However pipe sizes are much larger because of the large volume handled. These have horizontally split casing. Those having bottom connections are elevated. Concrete columns and tabletop arrangements are provided with surrounding access platform.

Compressor foundations are kept independent of building foundations. A hand operated travelling crane located at compressor centre line is to be provided overhead. Lay down space also has to be planned for. Knockout drums and inter stage exchangers are so arranged as to have short and simple piping. Long radius elbows should be used immediately before the compressor suction. A straight length of 5D at the suction is normally provided. For air compressors, strainer has to be provided at the open-air inlet.

Lubricating oil and seal oil consoles are also to be located which occupy large area near or under the compressors. Manufacturer's recommendation should be followed in this case.

The reciprocating compressors are located on concrete foundation at grade level and not on a concrete tabletop. The basis of lay down area, foundation and the roof above are identical to that of a centrifugal machine.

The piping interconnecting the pulsation dampeners, intercoolers, after coolers etc. should be short and simple to reduce the vibration. It is advisable to provide a vibration dampener (expansion bellow) at the outlet of the compressor. Sufficient straight length shall be provided after the discharge to avoid backpressure. To avoid vibration due to pulsating flow the customary piping details are modified as follows.

a) Use bends instead of elbows
b) Use angular inlets instead of laterals
c) Use smooth junction instead of head on opposing flow.
d) Use streamlined end-of-header arrangement instead of dead end header (Refer Fig 3.1.6). Components having large pressure drops should be avoided. Types of valves should also be selected accordingly. Support the piping at directional changes and at valves. Support should also restrict the pipe movements.

High pressure and high-speed compressors must have pulsation dampeners to eliminate pulsation in suction and discharge piping. If pulsating flow is transmitted to piping, structures and process equipment, material fatigue can occur. Dampeners' design is highly specialized and manufacturers recommendation should be followed. Nozzle velocities at inlet and outlet are limited to 15m/sec. While planning the layout, provision shall be made for these dampeners.

3.3 Heat Exchangers

The information required for the Equipment and Piping layout as applied to heat exchangers is the same as that required for any other equipment and has been explained earlier.

3.3.1 EQUIPMENT LAYOUT

The position of an exchanger in a Chemical or Petrochemical plant depends on the location of the distillation vessels/columns. The relative position should be evaluated from the process flow diagram (PFD).

The following general concepts apply for locating the heat exchangers.

a) Exchangers should be located adjacent to the related equipment, e.g. Reboilers should be located attached/ next to their respective towers, condensers should be located next to reflux drums close to tower.

b) Exchangers should be close to the other process equipment e.g. in case of draw off flow through an exchanger from a vessel/reactor bottom, the exchanger should be close to and under the vessel or reactor to have short pump suction lines. Overhead condenser shall be placed above the reactor to have minimum horizontal piping.

c) Exchangers connecting two equipment, one on shell side and the other on the tube side, located at a distance, should be placed where two streams meet, and on that side of the yard where majority of related equipment is placed.

d) Exchangers between process equipment and the battery limit. e.g. product coolers, should be located near the battery limit to reduce pipe rum.

e) Stack those exchangers, which can be grouped together to simplify piping and save plot space.

f) Leave space and access around the exchanger flanges and heads, and tube bundle cleaning/pulling space in front and in line with the shell.

g) While locating exchangers in a row, arrange the saddle to have more economical overall (lined up or combined) foundation / structure design. Further, travelling gantry can be provided in such case to handle a row of exchangers.

h) The heat exchanger shall be located in the equipment layout with respect to the fixed saddle and the same is located closer to the head.

i) Outline the clearances and working space in the front and around both ends of the exchanger to facilitate shell cover and tube bundle removal as well as maintenance and cleaning.

j) The channel end shall face the roadside for convenience of tube removal and the shell cover the rack side.

The various clearances shall be as indicated in Fig. 3.3.1.

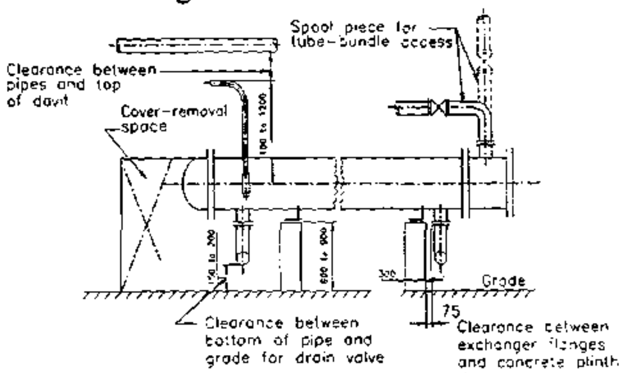


Fig. 3.3.1a

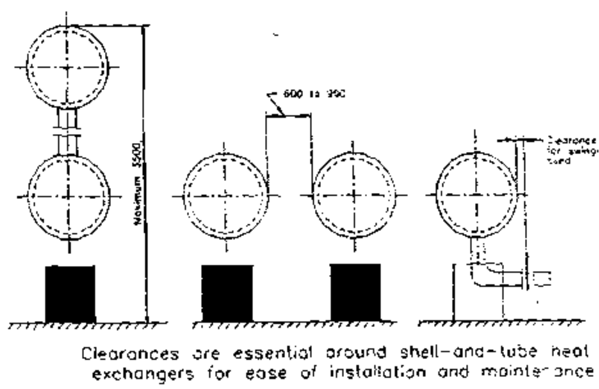


Fig. 3.3.1b

3.3.2 PIPING ARRANGEMENT

The piping engineer is responsible for the physical arrangement of piping. Most often the piping engineer will have no influence in the selection of exchangers, he can only modify the nozzle arrangements to have an economical piping. For this, the basic knowledge of the type of exchangers and its construction details are required. TEMA (Tubular Exchanger Manufacturers Association) specifies the various combinations possible.

The basic types used in the chemical process industry are

a) Fixed tube-sheet Heat Exchangers:

Exchangers with complete enclosed tubes are mainly used in clean services. Cleaning can be done by flushing through the tube side and shell side. Clean out connections are provided in the piping between the exchanger nozzle and block valve. The bolted cover and channel facilitate inspection and physical cleaning of the tubes. When used in high temperature services, an expansion joint is built into the shell to take care of the differential expansion between the shell and the tube.

b) 'U' tube Heat Exchangers:

In this type, the tube bundle is hairpin shaped and can freely expand. The bundle is removable from the shell and provision in layout has to be made for the same this type is used when fouling inside the tube is not expected.

c) Floating Head type Heat Exchangers:

This is more expensive than the fixed tube sheet and U-tube type exchangers. One end of the tube bundle has stationary tube sheet held between shell and channel flanges. The floating head can freely expand and contract with temperature changes.

d) Kettle type Heat Exchangers:

For high evaporation rates, kettle type heat exchangers are chosen. The shell is expanded to accommodate the generated vapor. This type generally has U-tube bundle.

The basic principles adopted in the heat exchanger piping are:

a) The working spaces should be kept clear of any piping and accessories to facilitate channel, shell-cover and tube bundle removal, as well as maintenance and cleaning.

b) Excessive piping strains on the exchanger nozzles from the actual weight of pipe and fittings and from forces of thermal expansion should be avoided.

c) The piping shall be arranged in such a way that no temporary support will be required for removing the channel and tube bundle.

d) Provide easily removable spool pieces, flanged elbows, break flanges, or short pipe runs to provide adequate clearances for the operation of tube removal.

e) The pipelines with valves and control valves should run along with access aisle close to the exchanger.

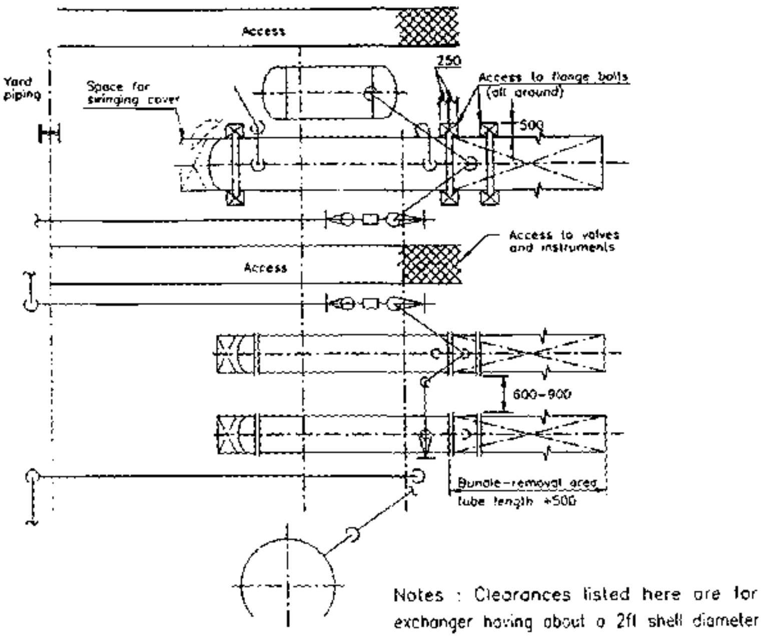
f) Pipe line connecting the exchanger with adjacent process equipment can run point to point just above required headroom.

g) Steam lines connecting the header on the rack can be arranged on either side of the exchanger.

h) Valve handles should be made accessible from the grade and from access way. This access way should be used for arranging manifolds, control valves stations and instruments.

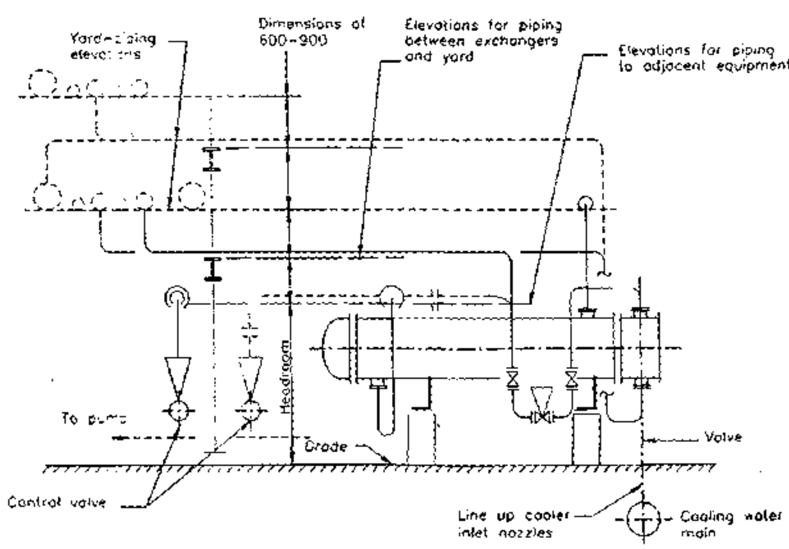
i) To avoid condensate drainage toward exchanger, the preferred connection for steam lines is to the top of the header. However, there is nothing wrong in having a steam connection from the bottom of the header if steam traps are placed at the low point.

j) The standard dimensions related to exchanger piping are given in Fig. 3.3.2.



Exchanger piping in plan shows arrangements for heat exchangers and space required for access

Fig. 3.3.2a



Exchanger piping in elevation shows location of pipeline runs in relation to main pipe rack

Fig. 3.3.2b

The following alterations can be suggested in order to achieve an optimum piping arrangement. Consent from the process group is required to ensure that these will not affect the thermal design of the exchanger. The cost increase in modification can be more than offset by the cost effective piping. These factors influence the decision on piping routing as well.

a) Elbow nozzle permits lowering of heat exchanger to grade to have better accessibility to valves and instruments. (Refer Fig. 3.3.3)

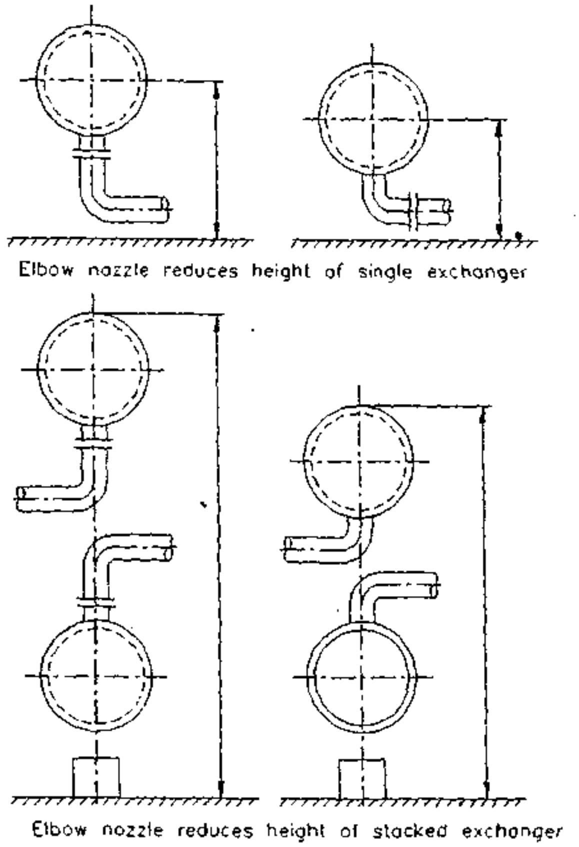
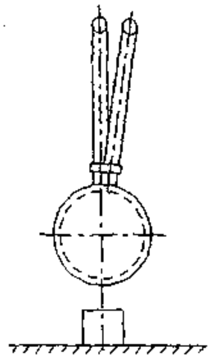


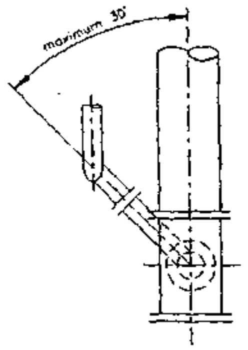
Fig. 3.3.3

b) Angular nozzle can save one or two bends in the pipeline. The maximum angle from the vertical centre line can be about 30^0. (Refer Fig. 3.3.4)



Angular connection for top nozzles

Fig. 3.3.4a



Angular connection for Bottom nozzles

FIG.3.3.4b

c) Horizontal exchanger can be turned vertical for conserving floor space. Vertical exchangers can be changed to horizontal when installation height is restricted.

d) Exchanger saddle can also be relocated to adjust to a line-up or combined foundation design. (Refer Fig. 3.3.2)

e) Interchange flow media between tube side and shell side. This can give the following advantages.

i) If hotter liquid is allowed to flow through the tube, this will minimize the heat loss and/or avoid use of thicker shell insulation.

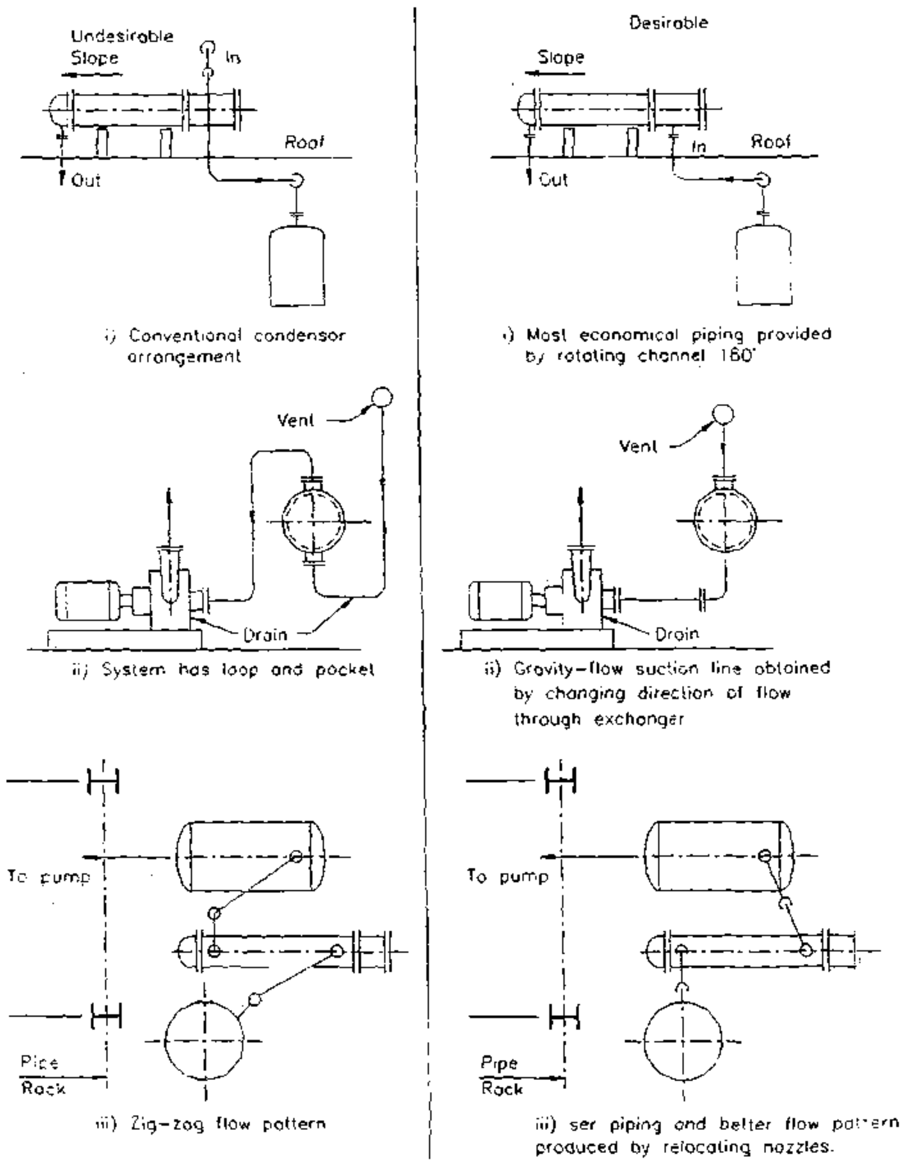
ii) If high pressure fluid flows on the tube side, only tubes, tube sheets, channels and

covers have to be designed for high pressure. This reduces shell side thickness and the cost.

iii) Corrosive liquid should pass through the tube so that only the tubes and the channels have to be made of corrosion resistant material.

iv) If one medium is dirty and the other is clean, passing clean through the shell will result in easier tube bundle removal and cleaning.

v) Shell side volume is much more than the tube side and hence vaporization or condensation of free flowing fluid is more effective in shell.



Simplifying the flow path improves piping design
Fig. 3.3.5

vi) When hazardous chemicals are water cooled, the water is passed through the shell. The tube leakage will contaminate the cooling water. On the other hand, the shell leakage can vent process material to the atmosphere.

Simplifying the flow path to improve the piping design is illustrated in Fig. 3.3.5.

3.4 Process And Storage Vessels

The basic set of information required for the equipment and piping arrangement for the process and storage vessel does not differ from those of other equipment. Design methods and conceptual details also differ very little. The process vessels can be classified, based on their function, as follows:

3.4.1 a) Surge volume to hold liquid for a specific length of time and

b) Liquid-vapor separation, or separation of immiscible liquids with different specific gravities.

This category include reflux drums, surge drums, process liquid collection drums, drums for additives, decanters, steam flash and condensate collection drums; caustics and acids holding drums etc. These could be horizontal or vertical.

3.4.2 Drums are with internals, often agitators, for mixing operations. These can be simple reactors with agitators of required type or with cooling/heating coils or jacket. The material of construction can vary from carbon steel, stainless steel and glass lined. These reactors are normally vertical.

These vessels are located in a process flow sequence. The area required around reactors is much more compared to other process vessels.

The manual loading of these, if required, also shall be facilitated. Hence these reactors will be located passing through the operating floors with the manholes located at accessible levels. Space for utility manifolds and process inlet manifolds, if required, shall be provided for in the layout.

3.4.3 Storage vessels and tanks can be in two categories:

a) Intermediate storage, generally located adjacent to process units or buildings.

and

b) Feed chemical or product tanks remotely located, the area being identified as tank farm with its dyking and acid proofing requirements.

These tanks are conical roof atmospheric storage tanks to store bulk chemicals. If the storage demands high pressure, these are designed as horizontal bullets or spheres. The relevant statutory requirements govern the layout of these storage tanks. These aspects are already covered in the 'Development of Plot Plan'.

3.4.4 PIPING ARRANGEMENT

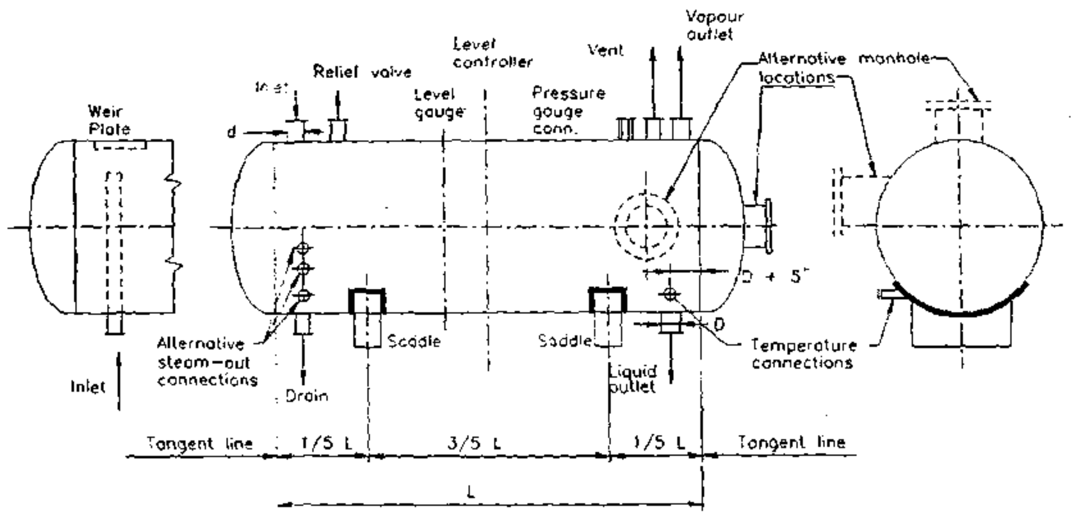
The piping associated with these vessels is simple. Economy of piping and access to valves and instruments depend on well-oriented nozzles. The nozzle and support orientation can be evaluated as below. (Refer Fig. 3.4.1)

a) Inlet/outlet nozzles

Vapor/liquid inlet is placed on top at one end. Bottom inlet is also possible but with a standpipe. Outlet is placed at the bottom on the other end. The bottom inlet is provided in case of large diameter piping to save pipe and fittings. In some cases, the inlet and outlet are centrally located. The vapor outlet in this case shall be diametrically opposite to liquid inlet.

b) Vents and Drains

Vent nozzles are located at the top and drain at the bottom of the other end.



Nozzle and manhole locations
Fig. 3.4.1

Vessels are sloped towards drain point. If the vessel has a top manhole, the vent can be located on the manhole cover. The drain valve can also be located at the low point of the outlet piping instead of vessel.

c) Relief Valves/Rupture Disc

Located anywhere on the top of the drum, preferably at accessible location from the platform provided for valve operation.

d) Level gauges

The gauge glass shall be located at the least agitated liquid section. If the vessel is horizontal, the location is best at the centre of inlet and outlet nozzle.

e) Pressure and Temperature tap-offs.

Pressure connection at the vapor space at the top of the vessels will make it visible from the platform. The temperature connection shall be near the outlet pointing towards the platform.

f) Manholes

These can be positioned on the top, to the side or at one end of the vessel. Depending upon the vessel size, there can be two manholes, one located on top and the other is located on the shell accessible from the grade.

g) Vessel saddles

Ideally, saddles are located at 1/5 of the drum length from each tangent line. The vertical vessel can be supported on skirt, on half skirt, on legs or on lugs as the case may be. The analysis of the support attachment shall be done accordingly.

Location of associated equipment around the vessel also influences the nozzle orientation. The liquid outlet should be located towards the nearby pump suction. The elevation of the vessel should be such as to provide the required NPSH. In a reactor, the manhole should be oriented towards the access aisle to facilitate manual feed if required. The utility manifold and the process inlet manifold should be placed at the operating level irrespective of the nozzle attached to the vessel.

The piping should run overhead as short and as simple as possible. Pipe rack shall be arranged which will support electric and instrument trays along with pipelines. Piping should not transmit vibrations and should be isolated from vibrating equipment with the help of expansion

bellows or hoses. Such equipments are centrifuges, filters, dryers etc. In all these cases, interaction with the manufacturers of equipment will always yield positive results.

While detailing the layout and piping, it should always be borne in mind that this should achieve the best performance as envisaged and detailed in the P & I Diagrams.

The structural arrangement for support also contributes a considerable part in the economy of installation. Certain economical arrangement of support is illustrated in Fig. 3.4.2 to 3.4.5, which are self-explanatory.

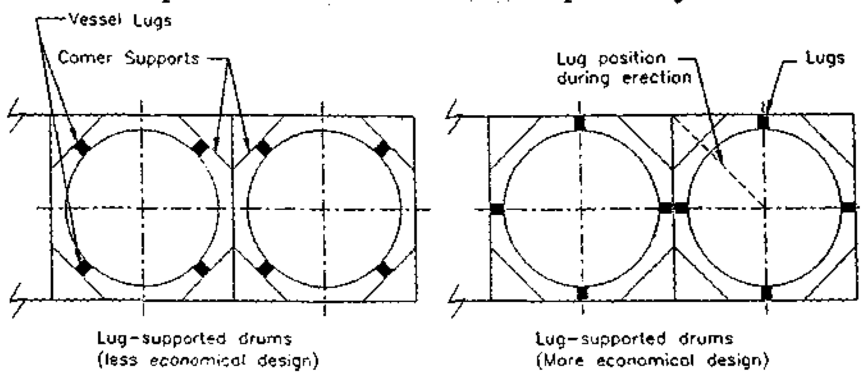


Fig. 3.4.2

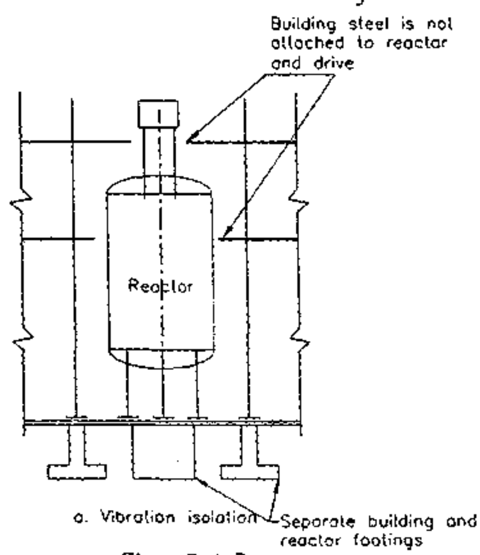


Fig. 3.4.3

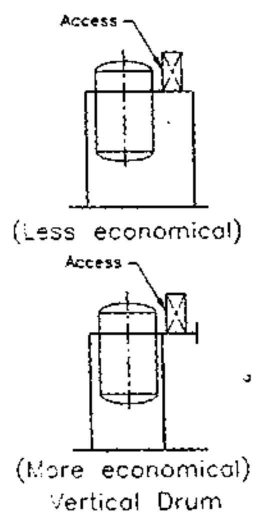


Fig. 3.4.5

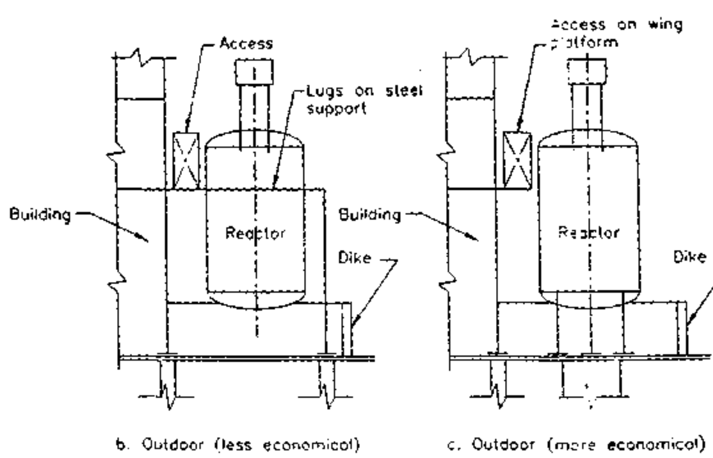


Fig. 3.4.4

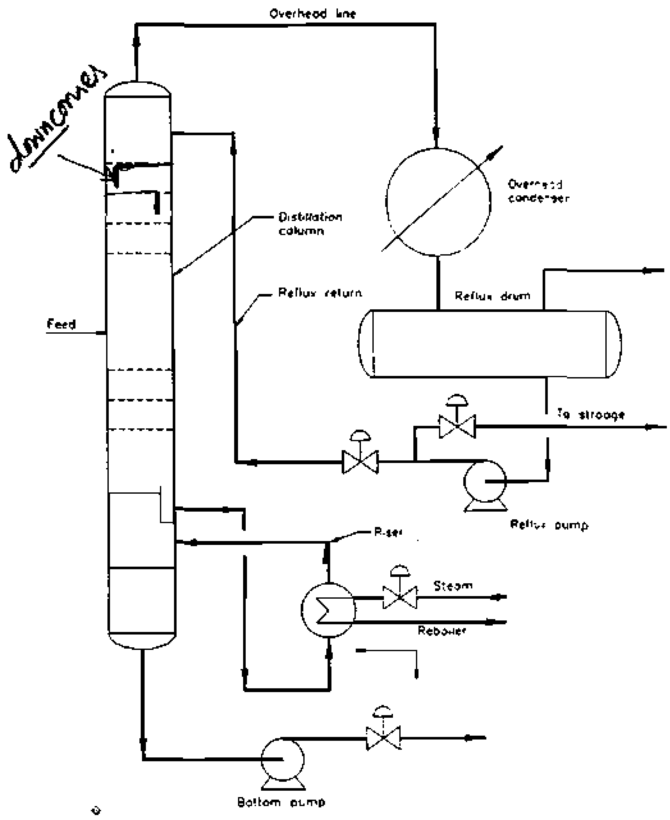
3.5 Distillation Units

3.5.1 BASICS:

Let us analyze the equipment layout and Piping design for a distillation column, which is more of an integrated unit than the individual equipment discussed earlier.

Interactions between hydraulic requirements and piping configurations require close attention to many fluid and mechanical details, in order to obtain the most efficient and economical distillation units.

The PFD of a typical distillation column with bottom pump, thermosyphon reboiler, overhead condenser, reflux drum are as shown in Fig. 3.5.1. During the normal operation, the pump transports the liquid in equilibrium. The arrangements should be such that the NPSH requirements of the pumps are satisfied. The pump normally has dual service. The pump head requirements should be selected to suit both services. The pump can also have simultaneous operation. All alternatives should be investigated while selecting the reflux pump.



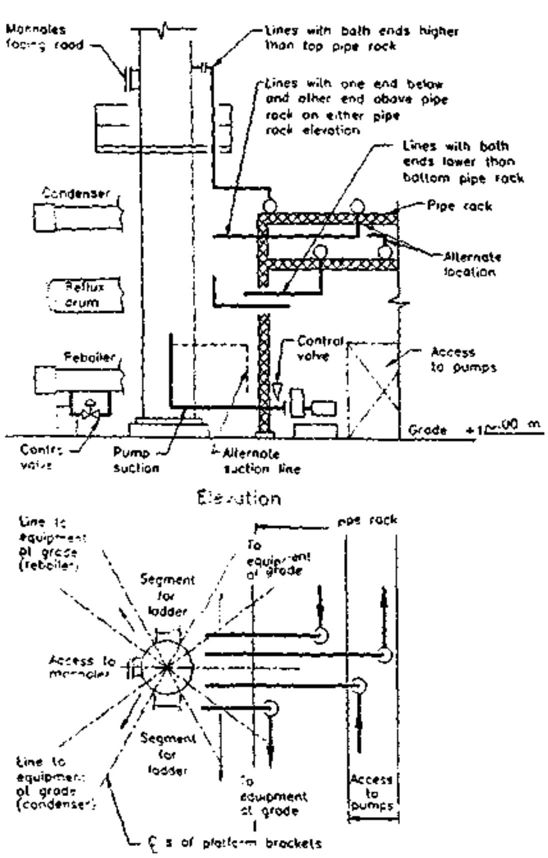
Process flow diagram
Fig. 3.5.1

The reboiler circuit could be either pumped or thermosyphon. The design of pumped reboiler circuit is similar to that of reflux pump system. The bottom pump transports 'the liquid through a heat exchanger and return to distillation column. The possible two phase flow in the pipeline after heater should be studied.

The reboilers could be shell and tube type, vertical or horizontal to kettle type. In large diameter towers inserted type U tube bundles inserted directly into liquid space through tower nozzles, are also used. Helical coils inside tower also can be used for small duties.

3.5.2 EQUIPMENT LAYOUT:

The equipment components are located adjacent to each other in a process plant. The arrangement of principal elements to integrate into the overall plant layout is as shown in Fig. 3.5.2.

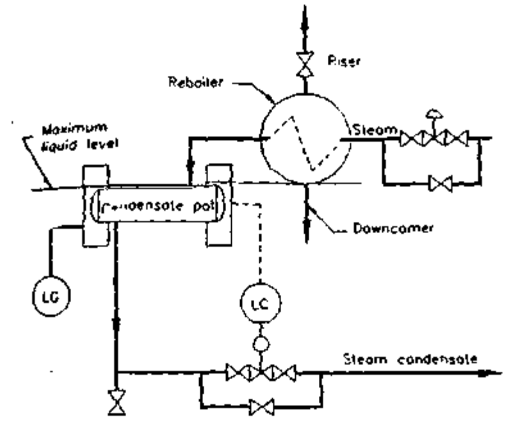


Piping around the distillation column:
Fig. 3.5.2

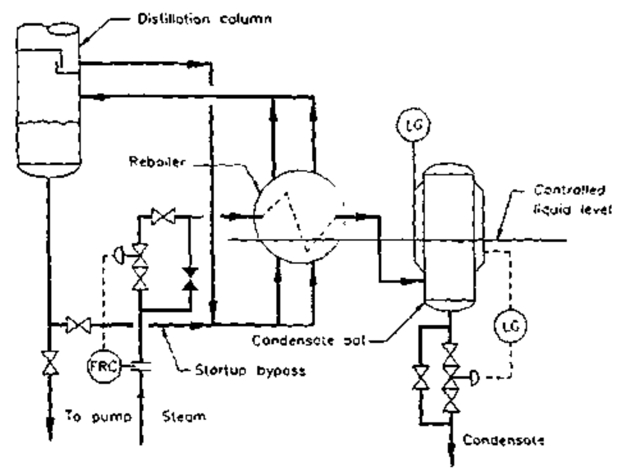
The tower is located adjacent to the rack so that the lines can drop directly on to the rack and can turn left or right. The plan view (Fig. 3.5.2) of the tower shows the segments of the tower along its circumference allotted to various utilities. Manhole faces the access roads or access aisles in case of indoor layout. Platform is provided below each manhole at a distance of 750 – 1000 mm below the centerline. This will facilitate maintenance as well as access to instruments and valves. From layout point of view, it is preferable to have equal platform spacing, and orientation of bracket lined up along with the entire length of the tower. This will minimize the interference with piping.

Most reboilers are at grade next to tower with centerline elevations 1.5 m to 2 m above ground. This is to facilitate tube bundle removal, maintenance access to valve and instruments. In this arrangement the static head is well determined between the exchanger centerline and draw off and return nozzles on the tower. Vertical reboilers are supported on the distillation column or adjacent to it at the same elevation. The details of support on the skirt or on adjacent structure are worked out considering the temperature of operation. Some reboilers have condensate or liquid holding pot located after the tube side outlet as shown. When high capacity steam traps are provided, the top of the condenser pot should not be higher than the bottom of the exchanger shell, to avoid flooding on the tubes with condensate adversely affecting the performance of the exchanger. Process conditions determine the precise relationship between the exchanger and the vertical condensate control pot as shown in Fig. 3.5.3.

Fig. 3.5.4 shows the part P & I diagram for a kettle type reboiler. This arrangement in the reboiler is elevated to meet the NPSH requirement of the centrifugal pump. The elevated reboiler, in turn, raises the tower because the minimum liquid level in the bottom of the tower must be higher than the liquid level in the heat exchanger. The elevation difference marked as H in the Figure provides static head for flow in reboiler circuit, and overcomes friction losses in the exchanger, down comer and return lines. The pumps are located below the rack as per the details indicated in section for pumps.

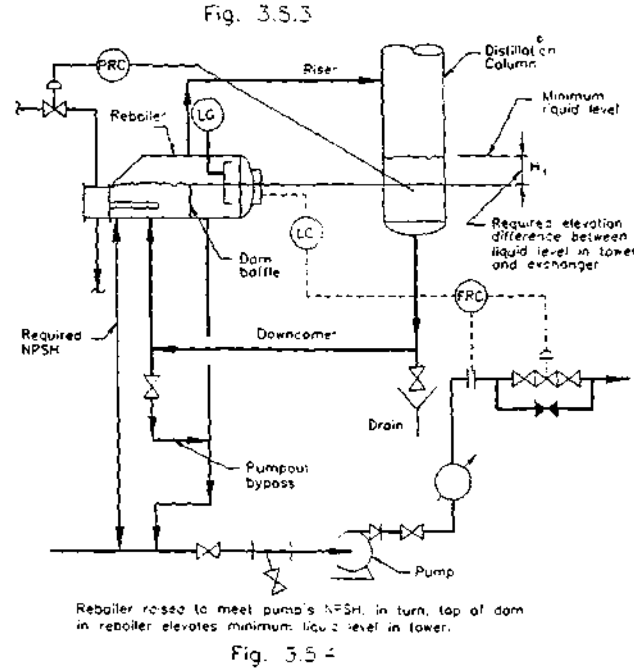


a. Bottom of reboiler should be elevated just above top of condensate pot.



b. Condensate pot regulates liquid level in exchanger tubes. Physical relationship between liquid level in condensate pot and required liquid level in exchanger tubes is important.

Fig. 3.5.3



Reboiler raised to meet pump's NPSH, in turn, top of dam in reboiler elevates minimum liquid level in tower.

Fig. 3.5.4

3.5.3 PIPING LAYOUT:

The segment used to locate the down comer from the column is as indicated in Fig. 3.5.2. The area segments of piping going to equipment at grade are available between ladders and on both sides of manholes. For economy and easy support, piping should drop immediately upon leaving the tower nozzle, and run parallel and as close to the tower as possible. The insulation thickness of tower and that of any possible flange on the tower as well as line should be considered while deciding the distance. The vertical line can be a suitable location for the straight run for an orifice.

The horizontal elevations after the lines leave the vertical run are governed by the elevations of the main pipe rack. The lines to run further on the rack can approach the rack at a higher elevation considering the size of other horizontal lines and be placed on the top tier. These can turn 'left or right' depending on the plant overall arrangement. Lines running directly to equipment at grade, more or less in the direction of pipe rack, often have the same elevation as the pipe bank.

Lines from tower nozzles below the pipe rack should approach the pipe bank below the rack elevation. The same elevation is used for lines that run for pumps located below rack.

The pump suction lines should be as short as possible and run without any loops or pockets. The steam lines to reboiler are tapped from the top of the main header to avoid excessive condensate drainage to process equipment. This line can run at the same elevation as those lines approaching the rack from the tower.

****

On distillation column the largest lines are the overhead vapor lines and reboiler down comer and return. These lines should have simplest and most direct configurations to minimize pressure loss and cost.

The prime consideration in all these cases is the performance to achieve the process requirements integrated with economy.

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TRANSIENT FLUID FLOW

Prof. A. S. Moharir
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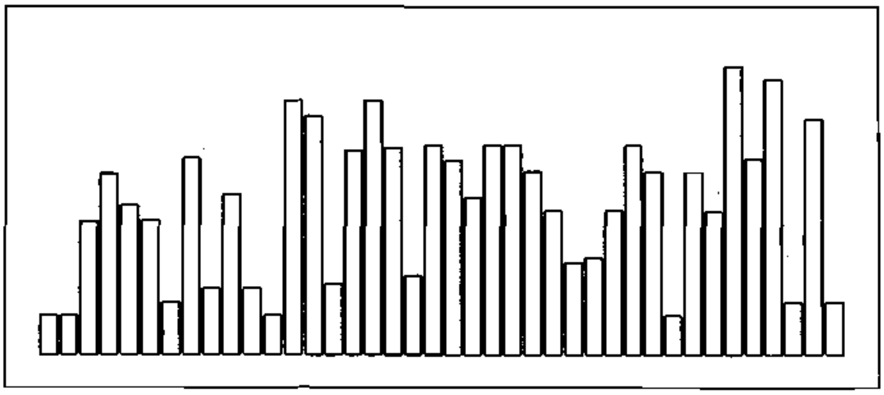

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TRANSIENT FLUID FLOW ANALYSIS

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Piping system's importance in any process plant cannot be over emphasized. In terms of cost, piping cost is estimated at approximately 25% of the total plant cost. This is next only to the equipment cost (approximately 50%). In terms of engineering man-hours, piping consumes almost half of the total man-hours. Same is the case for field man-hours spent on fabrication, assembly and testing. The operating bill is also bloated significantly by the energy consumed in transporting process or utility fluids through pipes connecting equipment. In transport and distribution business such as cross-country pipelines or gas networks etc, the entire capital and operating expenditure is in laying and running the pipelines. Piping systems, wherever and for whatever these may be employed, thus have a major footprint on overall project economics.

Wellbeing of piping systems is equally important for smooth and as-expected functioning of a process plant, just as it is in the case of process equipment. Pipes distinguish themselves as different from equipment, in terms of their aspect ratio (length to diameter ratio). A significant aspect is their length and rather complex routing which make them rather delicate and susceptible to failure or mal-functioning under single/multiple foreseen/unforeseen loads. Although design formulae for pipes and cylindrical process vessel are same as far as ensuring their integrity against internal/external pressures at sub- or super-ambient temperatures are concerned, process vessels are more stable and rigid compared to pipes due to their larger base (diameter) and unrestricted nature, at least in one dimension. Piping systems on the other hand, are 'anchored' at terminal, held/guided along the way by supports/hangars, must carry rather large weight en-route (valves etc.), move in complex 3-D configuration, undergo pronounced net expansion/contraction at operating temperatures different than installation temperature, must withstand vibrational and displacement loads transmitted by connecting equipment, must successfully negotiate occassional loads such as seismic wind, surge etc., must withstand flow induced vibrations and gravitational force etc. To foresee all these loads and select a pipe route exercises an engineer's knowledge in diverse science and engineering disciplines. That makes the so called 'piping engineer' so unique in an engineering organization.

This module deals with fluid transients. The origin of fluid transience, i.e. dynamic change in pressure and velocity profiles along a pipe route, otherwise designed for steady state behaviour, impact of these transient values on pipe integrity and methods to mitigate harmful fluid transients are discussed qualitatively as well as quantitatively.

Basics of Steady State Pipe Hydraulics

Mechanics is a branch of science which studies behaviour of objects under the application of external forces. Two fundamentally different forces are 'shear' and 'normal' forces. Normal forces could be tensile or compressive. Behaviour of a substances or an object under shear force is used to categorise matter, namely, solid and

fluid. A fluid is defined as a substance which when in static equilibrium cannot sustain shear forces. The term 'fluid' encompasses 'liquids' and 'gases'.

Compressive forces result in pressure. Fluid pressure is an important design parameter for piping systems.

One of the fundamental equations governing fluids in motion is the Bernoulli's equation. It is a statement of the law of conservation of energy as applied to fluids in motion.

Bernoulli's Equation

It is merely a statement of conservation of energy as applied to fluids in motion. It considers any fluid as having three types of energies, namely, potential energy, kinetic energy and internal energy (by way of fluid pressure). While allowing exchange of energy amongst these forms, Bernoulli's equation states that the total energy remains constant. It is best understood by considering a conduit carrying fluid between two points '1' and '2' as in figure 1. If P_i, Z_i and v_i; i =1, 2 represents the fluid pressure, elevation and fluid velocity at point i, then

$$Z_1 + \frac{P_1}{\rho g} + \frac{v_1^2}{2g} = Z_2 + \frac{P_2}{\rho g} + \frac{v_2^2}{2g}$$

The equation is valid for any pair of points along a conduit.

A special case is worth considering. Let the conduit be horizontal ($Z_1 = Z_2$) and of uniform cross-section ($v_1 = v_2$). The above statement of conservation of energy would then imply that the pressure is same at the two ends of the pipe. This is contrary to expectations as one expects pressure at upstream point ('1') to be higher than the pressure at the downstream point ('2') and this pressure difference actually sustaining the fluid flow. A force $(P_1 - P_2)\phi$ is then seen as acting on the fluid between points 1 and 2 causing it to move in direction $1 \longrightarrow 2$.

This is incorporated in the Bernoullli's equation by a 'loss' terms as follows:

$$Z_1 + \frac{P_1}{\rho g} + \frac{v_1^2}{2g} = Z_2 + \frac{P_2}{\rho g} + \frac{v_2^2}{2g} + h_f$$

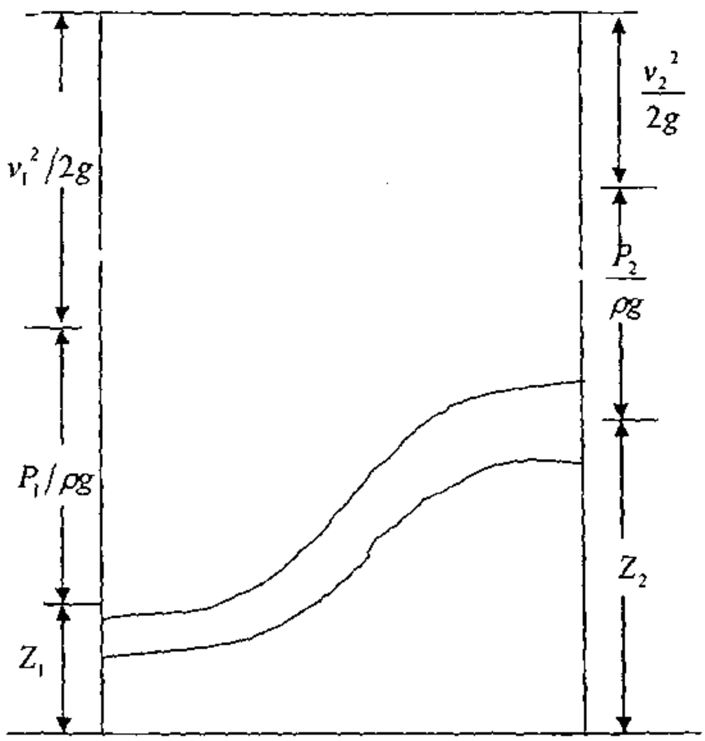
(for SI units, use g in P/ρg term)

h_f is the head loss which accounts for the loss of energy due to fluid friction and/or turbulence. In fact, this energy gets converted into heat, marginally increasing the fluid temperature which gets dissipated and is 'lost' for all practical purposes as it cannot be employed to do any useful work. Pumps and compressors are provided in the process plant essentially to sustain this 'loss'. This 'loss' constitutes a significant portion of operating cost. This constituent of the operating cost can be reduced by opting for a larger bore pipe for a given service. This, however, increases the capital investment. Pipe sizing is thus a balance between capital and operating costs. Reliable estimate of frictional

losses is thus important from process economics point of view. Some basic concepts and calculations are covered here.

Frictional Losses

Consider the earlier case of a horizontal, constant cross-section segment of a pipe transporting fluid from point 1 to 2. Let the pipe segment be straight and of length L. In addition, let the pipe be of circular cross-section with inner diameter 'd', let the steady velocity of fluid be v, the fluid density be ρ and viscosity μ.



Bernoulli's equation as applied to this straight pipe is:

$$\frac{P_1}{\rho g} - \frac{P_2}{\rho g} = h_f$$

$$\text{or} \quad (P_1 - P_2) = h_f \rho g$$

The fluid plug within the pipe between the points 1 and 2 experiences a differential pressure at its two ends. This causes a forces acting on the plug which is equal to

$$(P_1 - P_2)\phi$$

Where ϕ is the pipe cross-sectional area $(= \frac{\pi d^2}{4})$.

From above, this force is thus

$$h_f \rho g \quad \frac{\pi d^2}{4}$$

However, since the fluid plug is moving at a constant velocity (i.e. no acceleration), there can be no net force acting on the plug. The above force due to pressure differential is seen as negated by an equal and opposite frictional force between the fluid and the pipe wall in contact with the fluid. If this shear force per unit contact area (shear stress) is denoted as τ_f, one can write an overall force balance as

$$h_f \rho g \frac{\pi d^2}{4} = \tau_f \pi d L$$

The shear stress depends on fluid velocity (v), fluid properties (ρ, μ), and pipe properties ($d, \in$). $\in$ is a measure of pipe roughness and is expressed as an average height of 'roughness' mounds on the pipe surface. It is also called as 'equivalent sand roughness'. Thus

$$\tau_f = \tau_f (v, d, \in, \rho, \mu)$$

One can apply dimensional analysis to this qualitative functionality and arrive at the following relationship in dimensionless numbers.

$$\frac{\tau_f}{\rho v^2} = f\left(\frac{dv\rho}{\mu}, \frac{\in}{d}\right)$$

(For annular section, ∈ ... constant ... may be same as ...)

For a given flow of a given fluid in a given pipe, one can thus write

$$\frac{\tau_f}{\rho v^2} = \text{constant} = \text{X (say)}$$

$$\text{or} \qquad \tau_f = X \rho v^2$$

Using this in the force balance written earlier one gets

$$h_f = \frac{4XLv^2}{gd} = \frac{8XLv^2}{2gd}$$

This in the fundamental relation used to calculate frictional head loss for flow through pipes. X or its multiples are termed as friction factors (f). Three common definitions of the friction factor are as follows:

I. Churchil $\qquad f_C = X \Rightarrow h_f = \dfrac{8 f_C L v^2}{2gd}$

II. Fanning $\quad f_F = 2X \Rightarrow h_f = \dfrac{4 f_F L v^2}{2gd}$

III. Darcy $\quad f_D = 8X \Rightarrow h_f = \dfrac{f_D L v^2}{2gd}$

The use of three friction factors (without a subscript) is a source of confusion and often calls for care on the part of the user. Over- or under – estimation of pressure drop can be caused leading to erroneous pipe sizing or rating of pumps/compressors etc. if this point is overlooked.

The friction factor (f_C, f_F or f_D) is a function of two dimensionless numbers, Reynold's number ($dv\rho/\mu$) and roughness factor ($\in/d$). The dependence is normally presented graphically as friction factor vs Reynold's number plots with roughness factor as a parameter. Familiarity with these f vs R_e plots and a quick identification of which friction factor a given plot refers to is very important for all hydraulic system rating and design.

Several friction factor-Reynold's number correlations are also proposed as substitutes for the f - R_e plots. Some for Darcy's friction factor are collected below.

- Colebrook – White equation for turbulent flow

$$\frac{1}{\sqrt{f_D}} = -2 \log_{10}\left(\frac{\in}{3.71d} + \frac{2.51}{R_e\sqrt{f_D}}\right)$$

- For Smooth pipes

$$\frac{1}{\sqrt{f_D}} = 2 \log_{10}\left(\frac{R_e\sqrt{f_D}}{2.51}\right)$$

$$= 2 \log_{10}\left(R_e\sqrt{f_D}\right) - 0.8$$

- Blasius Equation

$$f_D = \frac{0.316}{R_e^{0.25}} \qquad \text{for } R_e > 10^5$$

- For rough turbulent flow

$$\frac{1}{\sqrt{f_D}} = 2 \log_{10}\left(\frac{3.71d}{\in}\right)$$

The above procedure is applicable only for straight pipelines. The actual pipe routing would have several piping elements (elbows, tees, expanders, reducers, etc.)

along its route as well as regulation, control valves etc. These cause extra turbulence and energy loss. For simplicity of engineering calculations each element is assigned equivalent length. This is the straight pipe length which would cause same pressure drop as the element for same flow rate. The piping element/valve can thus be replaced in

calculation by the equivalent (hypothetical) length. Total effective length of a pipe route is thus calculated. Frictional pressure drop for a given route is thus calculable.

Some Examples:

- Pressure drop in straight horizontal pipe
- Pressure drop in pipe with fittings
- Pressure drop in a complex pipe route
- Pipe in series, parallel, series-parallel
- Pipe Network
- Reserviors and pumps in network

Unsteady Flow Analysis

Steady state analysis (pressure drop due to fluid friction, Bernoulli's equation) can be applied to a complex pipe route or network to arrive at pressure and velocity at each node. A node is where there is a change of direction and/or diameter and/or flow. Normally, one would define a node each on either side of a straight, uniform cross-section pipe segment as well as on either side of a piping element or valve.

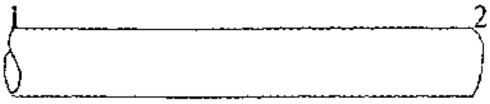
A steady state can be sustained as long as there is no external disturbance which momentarily or over a stretched time interval perturbs the flow conditions (flow rate, pressure, flow resistance at one or more nodes). The perturbation could be temporary and the conditions at the node return to their original status. In this case, the initial system at a steady state will deviate from it, but return to same steady state condition after sufficient time. The transient behaviour of the system from the point of application of disturbance to return to steady state is of interest in transient analysis. In other cases, the deviation may be permanent. The system will then reach another steady state after going through a transient phase. The transient phase is thus straddled by steady state on either side. These steady states serve as initial/ boundary conditions in the transient analysis. The steady state analysis is thus important from the point view of transient analysis as well.

The transient phase could take the system monotonically from one steady state to another or take the system through over-,under-, or critically damped oscillations. Some simplified situation could be analysed to appreciate the mathematics and behaviour of fluid transients.

In the following examples, the fluid has been assumed to be incompressible. Also, the changes in pressure are assumed not to cause change of phase. For example, consider a liquid in the piping system well above its vapour pressure throughout.

Example 1 :

Consider a straight horizontal pipe as shown.



At steady state, a force balance over the pipe section between two points '1' and '2' was written earlier as

$$\phi(P_1 - P_2) = \frac{f_0 L v_0^2}{2gd}\phi\rho g$$

The term on the left hand side is force on the liquid plug due to pressure difference at its two ends. The term on the right hand side is the shear force (frictional force) between the moving fluid and the stationary pipe wall. 'φ' is the cross-sectional area of the pipe. v_0 is the steady state velocity.

If for some reason, say downstream pressure 'P_2' is suddenly changed at time zero to P_2^1 and then held constant at P_2^1 subsequently, the velocity will change with time and finally stabilize at the new steady state value v^1 given by

$$\phi(P_1 - P_2^1) = \frac{fLv^{12}}{2gd}\phi\rho g$$

The velocity will change from v_0 to v^1 during the transient phase and one would like to get a measure of that and also the governing mathematical model.

The fluid clearly experiences an acceleration (or deceleration) as it changes velocity from v_0 to v^1. There must, therefore, be a net force acting on the fluid. This is the difference between the force due to pressure difference [$= \phi(P_1 - P_2^1)$] and shear force at any time t when the velocity is v

$$\left[= \frac{fLv^2}{2gd}\cdot\phi\rho g\right].$$

The net force must be equal to mass of the fluid plug between points '1' and '2' multiplied by acceleration $\left(= dv/_{dt}\right)$ at that instant.

Expressed in an equation

$$\phi(P_1 - P_2^1) - \frac{fLv^2}{2gd}\phi\rho g = \rho\phi L\frac{dv}{dt}$$

This is a differential equation in v. This can be simplified as

$$\frac{dv}{dt} = \frac{P_1}{\rho L} - \frac{P_2}{\rho L}^{1} - \frac{1}{2d} \cdot fv^2$$

(Remember that the friction factor 'f' itself is a function of v). This nonlinear first order ordinary differentical equation is subject to the initial condition.

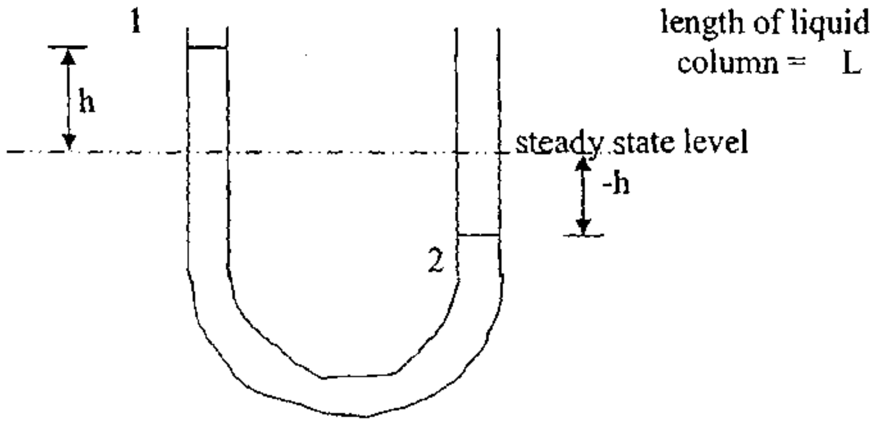
$$v = v_0 \quad \text{at} \quad t = 0$$

The solution gives a v vs time profile. Without solving the equation, one can qualitatively say here that the velocity will monotonically change from v_0 to v^1. The rate of change will be high initially and gradually taper.

One can extend this to a pipe which is not horizontal as well as a piping systems with several piping elements as well as valves. For a simple case of laminar flow, one could even attempt and get an analytical expression. Also the perturbation which sets up fluid transients could be due to different causes and of different temporal nature. Some such exercises can lead to a better appreciation of Bernoulli's equation extended to transient behaviour.

Oscillating Transience

An interesting case of how a perturbation can set up oscillations in a system is offered by a simple U tube. This is discussed below and a simplified mathematical model is developed and solved.



For simplicity, let us assume that there is negligible frictional loss as the fluid (manometric) moves through the limbs. Let the pressure be same in both the limbs. This would mean that the manometric fluid level in both the limbs is same. Let that level be the datum level.

Let at time zero, the column in the right limb be pushed down below the datum by height h_0 (thereby also raising the column in the left limb by h_0 above the datum) and

released. This perturbation sets up an interesting dynamics in the systems, which can be modeled as follows:

Let the fluid column be at position $\pm$h at time t as shown. Applying the Bernoulli's equation modified for transient behaviour earlier, we get

$$\frac{P_1}{\rho g}+\frac{v_1^2}{2g}+h = \frac{P_2}{\rho g}+\frac{v_2^2}{2g}-h+\frac{L}{g}\frac{dv}{dt}$$

As $P_1 = P_2$ and $v_1 = v_2 = v$

$$h = -h+\frac{L}{g}\frac{dv}{dt}$$

Also $v = \frac{dh}{dt} \Rightarrow \frac{dv}{dt} = \frac{d^2h}{dt^2}$

Therefore, the equation governing the dynamics is

$$\frac{d^2h}{dt^2}+\frac{2gh}{L} = 0$$

If we guess the solution as

$$h\,(t) = c_1 \cos c_2 t$$

we get $c_1 = h_0$

and $c_2 = \sqrt{2g/L}$

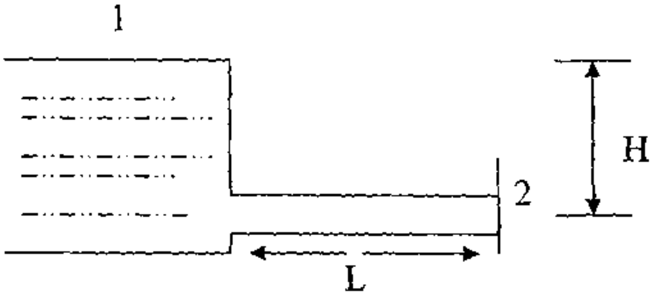
We thus observe the liquid column oscillating with an amplitude of h_0 and a frequency of $\sqrt{2g/L}$ radians/s. The frequency in cycles/s (i.e. Hz) is $\frac{1}{2\pi}\sqrt{2g/L} = \frac{1}{\pi}\sqrt{g/2L}$. The period of vibration is $\pi\sqrt{2L/g}$.

What happens if the frictional losses are not neglected ?

Can we make a statement as to when a perturbation can lead to this kind of oscillatory response in a system ?

Transient Discharge from a Tank

Consider a tank of a large diameter connected to a horizontal, uniform cross-section tube of length L. Let the water level in the tank be at a height H from the tube axis. Allow some water to run through the tube and then close the tube exit (Point 2). Top of the liquid in the tank is designated as point 1. Now the tube is full of liquid and so is the tank. At time zero, instantaneously open the tube exit to allow the liquid to flow. See figure for visualization. Develop and solve the mathematical model of this system.



At any time 't' after the tube exit at point '2' is opened, Bernoulli's equation modified for transient flow is :

$$\frac{P_1}{\rho g} + \frac{v_1^2}{2g} + Z_1 = \frac{P_2}{\rho g} + \frac{v_2^2}{2g} + Z_2 + h_f + \frac{L}{g}\frac{dv}{dt}$$

For the problem under consideration;

$$v_1 = 0$$
$$v_2 = v$$

$$P_1 = P_2$$
$$Z_1 - Z_2 = H$$
$$h_f = \frac{fLv^2}{2gd}$$

Therefore,

$$H = \frac{v^2}{2g} + \frac{fLv^2}{2gd} + \frac{L}{g}\frac{dv}{dt}$$

Let the velocity at steady state be v_0. Then, from steady state Bernoulli's equation,

$$H = \frac{v_0^2}{2g} + \frac{fLv_0^2}{2gd}$$

Subtracting the steady state equation from the unsteady state equation, one gets

$$\frac{L}{g}\frac{dv}{dt} = \frac{1}{2g}\left(1+\frac{fL}{d}\right)\left(v_0^{\,2}-v^2\right)$$

This is subject to the initial condition

$$v = 0 \quad \text{at} \quad t = 0$$

Rearranging and writing in integral form:

$$\int_0^v \frac{dv}{v_0^{\,2}-v^2} = \frac{1}{2L}\left(1+\frac{fL}{d}\right)\int_0^t dt$$

Integration using partial fractions gives

$$\frac{1}{2v_0}\ \ln\frac{v_0+v}{v_0-v} = \frac{1}{2L}\left(1+\frac{fL}{d}\right)t$$

OR

$$\frac{v_0+v}{v_0-v} = \exp\left[\frac{v_0}{L}\left(1+\frac{fL}{d}\right)t\right]$$

Abbreviating $\frac{v_0}{L}\left(1+\frac{fL}{d}\right)$ as c, one can write

$$\frac{v_0+v}{v_0-v} = \exp(ct)$$

or

$$\frac{v}{v_0} = \frac{e^{ct}-1}{e^{ct}+1}$$

One can thus see that the fluid velocity v reaches steady state velocity monotonically. Time to reach a certain percentage of value of the steady state velocity can be found out easily from above.

1. What is ct for $\frac{v}{v_0} = \frac{1}{2}$?
2. What is ct for $\frac{v}{v_0} = 0.99, 0.999$?
3. What mistake did we make in the above derivation?

4. Under what flow conditions, the mistake is not so serious?

5. What happens if flow is laminar throughout?

6. What happens if the tube is vertical and drawn from bottom of the tank?

7. What happens in the reverse case when the flow is established and then the exit is closed suddenly at $t = 0$. Stretch this as $L \to \infty$.

Water Hammer (Something is flowing & stopped, it creates the effect of water hammer)

Mulling over the last question leads to appreciation of water hammer.
For simplicity, let us consider that the liquid was incompressible and the pipe was rigid (non-elastic). Let the valve be closed over say a period of τ seconds after a steady state flow and with fluid velocity of v_0 has been reached. An estimate of the pressure that it would develop can be arrived at as follows.

The mass of fluid plug in the horizontal pipe (length L, cross-sectional area ϕ) is $\phi L\rho$. This has to be brought to a zero velocity from a velocity v_0 over a period of τ seconds. It thus needs to be imparted a deceleration (negative acceleration). Its magnitude can be found by using the instantaneous velocity vs acceleration formula.

$$v = v_0 + at$$

Where v is velocity at time 't' and 'a' is acceleration. For v to be zero at $t = \tau$, acceleration should be

$$a = -\frac{v_0}{\tau}$$

A mass (m) of fluid in the horizontal pipe $(m = \phi L\rho)$ can be imparted this acceleration (a) if a force F (= ma) acts on it in a direction opposite to the flow direction.

$$F = \phi L\rho\left(\frac{v_0}{\tau}\right)$$

Or

$$\frac{F}{\phi} = \frac{L\rho v_0}{\tau} = P$$ (formula for step wise [illegible])

F/ϕ is the pressure that would be generated at the exit end of the pipe. This pressure which is generated due to sudden closure of a valve is called water hammer. Water hammer effect can be severe if the length of the pipe upstream of the valve is larger, and/or the density of the liquid is large, and/or the velocity of the fluid prior to valve closure is high, and/or the valve closure time is small. Excessive pressure can lead to system failure. While L, ρ, v_0 are dependent on layout, service and pipe selection, the

closure time (τ) can be suitably planned through valve selection, design and operation. Valve choice is thus crucial from the point of view of mitigation of water hammer effects.

The above is a highly simplified picture of water hammer. It none the less provides a quick conservative estimate of pressure levels that could be expected to develop due to sudden valve closure. The actual pressure developed will be somewhat less than this estimate as the liquid is not totally incompressible nor is the pipe rigid. Compression of the liquid and elastic expansion of pipe cross-section due to heightened pressure help mitigate the water hammer effect to some extent.

Other complications are 3-D layout of piping systems in reality, to and fro travel of pressure wave and possibility of its reflection, thermodynamic behaviour of the flowing medium at different pressures which may cause phase change (evaporation, condensation). Applied mechanics, fluid mechanics and wave mechanics all get together to decide the pressure-velocity profiles with time and space in a piping system subject to sudden closure (or opening) of a valve or a failure, restarting of a pump. The mathematics is quite complicated and calls for numerical solutions of the governing partial differential equations with numerous boundary conditions. Competent s/w tools are available which expect the user to 'define' the system. Setting up of equations and boundary conditions as well as solving these to generate pressure-time-space and velocity-time-space profiles in the piping system is the job of the s/w. The user then analyzes the results and suggests remedial measures (surge vessel, pressure relief valve, valve closure pattern, control valves, etc.) to safeguard the piping system in the event of happenings leading to water hammer effects.

The following sections deal with water hammer analysis with the aim to develop a feel for physics so that educated use of such computer aided analysis is possible.

A simplistic formula was presented earlier to estimate pressure that could be generated by quick closure of a valve. For instantaneous closure ($\tau = 0$), it would estimate an infinitely large pressure rise. In reality, finite liquid compressibility and pipe elasticity prevent this infinite pressure rise. The following discussion pertains to making the estimate more and more realistic in view of actual fluid and pipe properties.

Finite Liquid Compressibility, Rigid Pipe

Joukowski formula gives a more realistic estimate of pressure rise as follows.

$$\Delta P = \rho c \Delta v$$ (water hammer for sudden closure)

Here, ΔP is the maximum expected pressure rise due to sudden valve closure, ρ is fluid density, Δv is the effected change in fluid velocity and c is the velocity of pressure wave in the medium. It is related to the fluid properties as follows.

$$c = \sqrt{E_f / \rho}$$

where E_f is the bulk modulus of elasticity of the fluid.

bulk modulus of elasticity = change in [illegible] / change in volume

for water

Consider for example a case where water in a pipe flowing at $2\,m/s$ velocity is brought to halt by closing a valve instantaneously.

$$E_f = 2.2 gPa.$$

Finite Liquid Compressibility, Elastic Pipe

Joukowski formula is applicable here as well. A composite modulus of elasticity (E_c) is defined in place of bulk modulus of elasticity of fluid as follows.

$$\frac{1}{E_c} = \frac{1}{E_f} + \frac{D_0}{tE_p}$$

E_f and E_p are modulus of elasticity for fluid and pipe respectively. E_c is the composite modulus of elasticity. E_c is used in place of E_f in calculating pressure wave velocity in the medium. An equivalent definition of this velocity can be shown to be as follows.

$$c = \frac{\sqrt{E_f\ \rho}}{\sqrt{1 + \frac{E_f}{E_p}\frac{D_O}{t}}}$$

The numerator is the pressure wave velocity in the fluid in a rigid pipe and the denominator can be viewed as a correction for pipe elasticity. D_O is pipe OD and t is pipe thickness.

One can thus generate a feel for the sensitivity of the water hammer effect.

Repeat the above calculation for a 4" SCH 40 pipe made of metal $\left(E_p = 207\times10^9\,Pa\right)$, as well as plastic pipe $\left(E_p = 1.4\times10^9\,Pa\right)$. See what happens if the pipe is 8" SCH 40 instead.

Why is the length of the pipe not involved in any of these calculations?

Multi Section Pipe

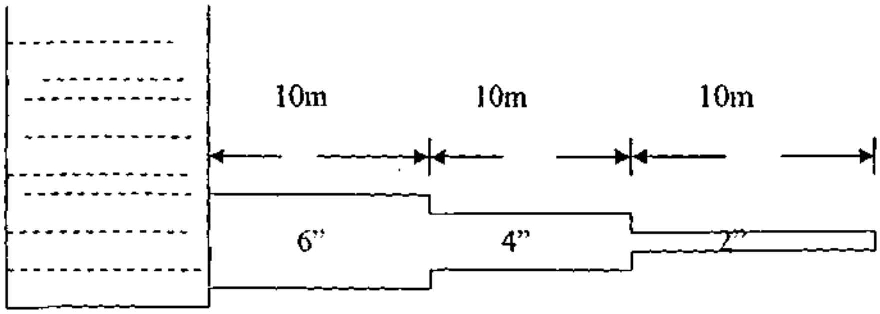
Consider the earlier example with the only change that the pipe is made of three sections of different diameters (say 6", 4", 2" SCH 40 pipes). Let the lengths be 10m each for each section and velocity be 2m/s in the 2" section. Repeat the calculations for the cases where liquid is incompressible and pipe is rigid, liquid is compressible and the pipe rigid, liquid and pipe are both elastic.

Sound velocity in water = 1056 m/s (at room temp.)

in air = 330 m/s

Sound velo. in metal is more than rubber pipe

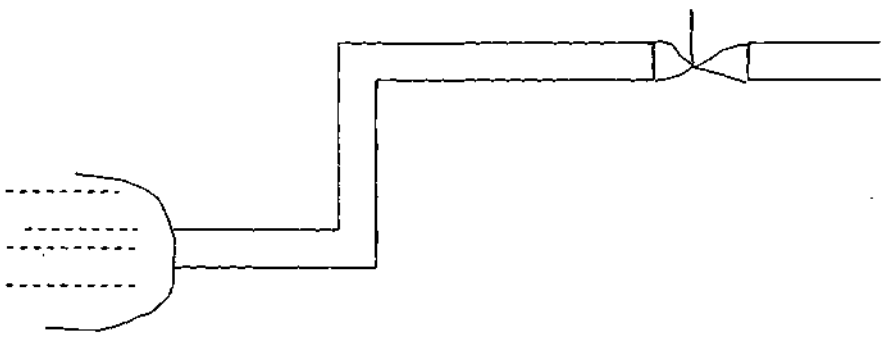
To take care of water hammer, expansion loop Joint can be used next to valve



What will happen if the telescope is reversed?

2-D, 3-D Pipe Routes

Consider the earlier case of an infinitely large reservoir connected to the discharge pipe. The only difference being the pipe runs in 2-D, still horizontally. A plan of the scheme is shown below.



In this case also, maximum pressure is generated at the point of application of sudden closure, i.e. at the valve end of the pipe. The pressure wave generated thus will travel upstream and hit the first elbow/bend. This would create a pounding effect and the wave will be reflected downwards. It will hit the next elbow/bend, create a pounding effect and get reflected towards the source. The source being a large, relatively stagnant mass would reflect the wave back. It will travel back, create pounding effect on the two elbows and reach the valve end. It will then get reflected again and so on. The pounding creates the audible water hammer effect. The fluid in the pipe also senses the local-temporal pressure gradients and moves helter-skelter. The pressure attenuates due to continuous loss of energy due to friction between fluid and pipe wall as well as minor losses in the piping elements (elbows, entry from tank to pipe). Rigorous water hammer calculations are done to generate these pressure and velocity profiles at important locations with time. The water hammer effect can also be seen as creating vibrations in a piping system. These need to be taken into account in dynamic analysis as occasional load.

The reflection of the pressure wave can be advantageous in the sense that it further reduces the peak pressure that can get generated in a system. To what extent the maximum pressure (as calculated by Joukowski formula) is affected by pressure wave reflection depends on the layout and the distance to be travelled from the peak pressure location to the point of possible reflection. Although the formula (Joukowski) is for instantaneous closure, real life valve (even if they fail closed unintentionally) take finite

time to close. Relation of the magnitude of this time to the time required for wave reflection is an important factor that decides the maximum pressure that gets generated.

Maximum pressure gets generated at the point of application of closure if the valve closes in a time (τ) less than the time taken for the pressure wave to start from the point of reflection, reach the point of application and return to the point of application. The returning wave can negate some of the pressure and reduce pressure levels.

If the distance between the point of closure and reflection is L, the above can be expressed as follows:

If $\tau \leq \frac{2L}{c}$ pressure surge is maximum.

If $\tau > \frac{2L}{c}$ pressure surge will reduce. (water hammer will reduce)

A reasonable correction for the maximum pressure surge given by Joukowski formula for cases where valve closure spans over a period (τ) more than $2L/c$ is given by

$$\Delta P = \left(\frac{2L/c}{\tau}\right)\rho c \Delta v$$

$$= \left(\frac{2L/c}{\tau}\right)\Delta P_{max}^{Joukowski}$$

The reflected and the original waves can cause periods and locations of high and low pressure. Periods of low pressure can cause degassing of liquids or even evaporation. Similarly, high pressures can cause re-condensation etc. High and low pressures also can cause periodicity of stresses in the pipe and eventual fatigue failure.

Fluid transients can occur due to several other reasons apart from sudden valve closure. The other reasons could be sudden opening of valves, starting or stopping of pumps, changing elevation of reservoir, change in power demand of turbine, reciprocating pumps, tube rupture in an heat exchanger, etc.

Anything that suddenly changes the steady state behaviour by changing one or more flow parameters sets up transient response. This could take the pressures to reach levels beyond the range confined by initial and eventual steady state. Measures need to be taken to ensure that these levels do not damage the systems.

Mathematical Model for Fluid Transients

Flow transience in any piping system or its section is captured in the following two simultaneous partial differential equations.

$$\frac{\partial P}{\partial t} + \rho c^2 \frac{\partial v}{\partial x} = 0$$ (mass is conserved)

$$\frac{\partial v}{\partial t} + \frac{1}{\rho}\frac{\partial P}{\partial x} + \frac{fv|v|}{2D} = 0$$ (momentum is conserved)

The two dependent variables are the pressure and the velocity of fluid. The two independent variables are time and space.

The equations can be solved with suitable initial/boundary conditions. These would be situation specific. For a simple case of a straight pipe discharging from a large reservior through a valve and the valve then closed over a period τ as per some closing curve, the conditions could be visualized as follows.

P (x = 0, t) = ρgh
P (x = L, t) = Pressure drop across valve
P (x, t = 0) = Steady state pressure
v (x, t = 0) = Steady state velocity
v (x = L, t) = Valve discharge rate/ pipe ϕ

For pipe routes with different diameters and other valves en-route, similar conditions may be required over each section.

These equations are solved with powerful solvers using method of characteristics, finite difference method, orthogonal collocation etc. Accuracy and convergence of the solution depends on suitable choice of space-time ($\Delta x - \Delta t$) grid. Choice of grid must observe certain logical restriction such as

$$\Delta x \geq c\Delta t$$

That grid size which gives grid-size-independent solution is desired. Finer the grid, more accurate could be the solution and larger the computational effort. One needs to strike a balance between accuracy and solution time.

It may be helpful to derive the basic model from first principles. No new concept is actually involved other than conservation of mass and energy. The equations are actually statements of mass conservation and Bernoulli's equation.

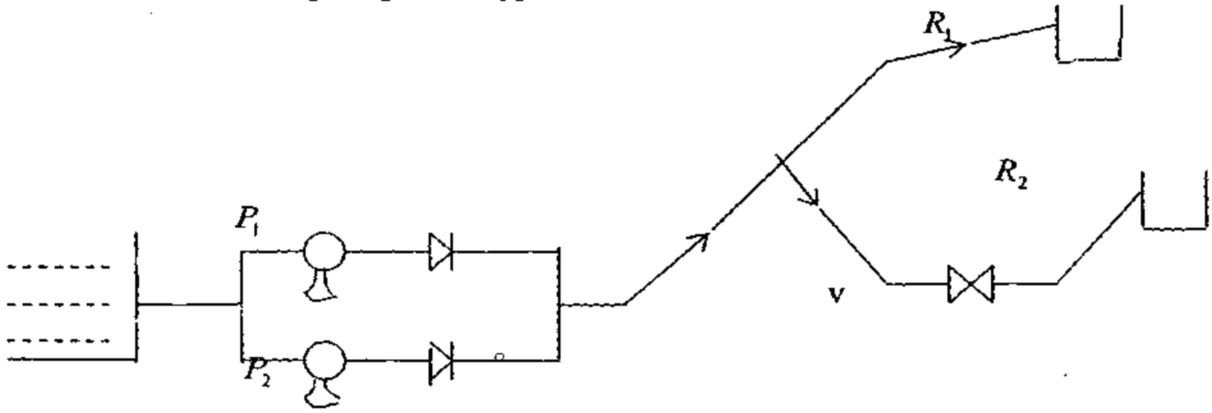
It is also useful to mull over the reasons for a product $v|v|$ appearing in one of the equations. What physical statement is one making in this term?

If due to pressure reduction, there is degassing of liquid or minor evaporation causing tiny bubbles to form and get dispersed in the liquid, what would be the effect on pressure surge? If pressure rise is unacceptably high, what are the possible remedies?

Pump and Valves

The occurrence of pressure surge has been discussed so far in relation to the sudden shutting down of the valve. In general, pressure surges would occur as a consequence of any event which causes rapid change in the velocity of the fluid. Rapid closing or opening of the valve or start or shut down of a pump could cause such a change in velocity and hence pressure surges. The role of pumps and valves in the piping system is discussed here further.

Consider the following simple but typical water distribution scenario.



Water is pumped from an underground reservoir to two high level reservoirs to facilitate supply against day-time non-uniform demand of customers. The pumps operate in parallel and simultaneously till tank R_2 (which is smaller in size as compared to R_1) is near full. The butterfly valve is then closed over a short period. Once the valve is fully closed, Pump P_2 also trips over a short period.

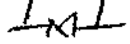
Comment on the possible happenings and pressure surges that could develop in the system for a rapid and not so rapid valve closure.

- Compile a list of data that you would collect to be ready for transient analysis.
- Discuss qualitatively worst case scenario (very rapid closure)
- Discuss instructions/ measures to avoid abnormal pressures.

Control Valves

If there is a pressure/ flow control valve on the line experiencing surge, the control valve dynamics would modify the surge characteristics. A knowledge of control valve characteristics and control law is important. Some basic background of the types of

control valves and control strategies as well as dynamics of valve-pipe systems is important for transient analysis.

- For creating passage for liquid (say water) provide bypass 

Ex: we can also provide nitrogen provided (gas provided) @ tank with liquid in it, If there is pressure drop then liquid may move from valve to tank. (In this case, we have to take care of reverse flow) (e.g. kerosine line)

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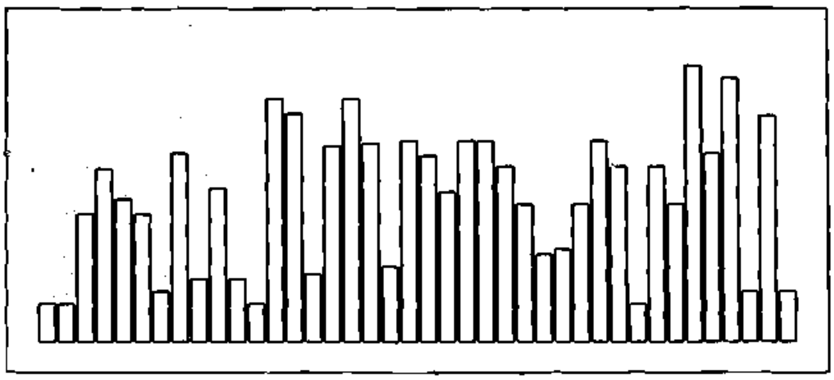
PIPING ENGINEERING

June 12 - 25, 2006

PIPE UNDER STRESS

Prof. A. S. Moharir
IIT Bombay



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PIPE UNDER STRESS

PROF. A. S. MOHARIR

INTRODUCTION

Pipes are the most delicate components in any process plant. They are also the most busy entities. They are subjected to almost all kinds of loads, intentional or unintentional. It is very important to take note of all potential loads that a piping system would encounter during operation as well as during other stages in the life cycle of a process plant. Ignoring any such load while designing, erecting, hydro testing, start-up, shutdown, normal operation, maintenance etc. can lead to inadequate design and engineering of a piping system. The system may fail on the first occurrence of this overlooked load. Failure of a piping system may trigger a Domino effect and cause a major disaster. This is the lesson from the infamous Flixborough disaster that everybody having anything to do with design, engineering, maintenance, operation etc. of a piping system must learn. It is not sufficient to do 99 right things and 1 wrong thing while designing a piping system. The end result would be disastrous. One must score a perfect 100 in piping system design.

The idea of this paper is to discuss all possible potential loads that are developed in a piping system and their implication on the stresses that would be generated in the pipes. Some guidelines to minimize the effect of such loads and keep the resultant stresses under limits specified by the codes are then given. Final design and engineering of a piping system may have to go through rigorous calculations, either manual or on computer, of the collective effect of all such loads and sound analytical skills to take engineering decisions to mitigate this effect.

Stress analysis and safe design normally require appreciation of several related concepts. An approximate list of the steps that would be involved is as follows.

1. Identify potential loads that would come on to the pipe or piping system during its entire life
2. Relate each one of these loads to the stresses and strains that would be developed in the crystals/grains of the Material of Construction (MoC) of the piping system.
3. Decide the worst three-dimensional stress state that the MoC can withstand without failure
4. Get the cumulative effect of all the potential loads on the 3-D stress scenario in the piping system under consideration.
5. Alter piping system design to ensure that the stress pattern is within failure limits.

The goal of quantification and analysis of pipe stresses is to provide safe design through the above steps. Of course, there could be several designs, which could be safe. A piping engineer would still have a lot of scope to choose from such alternatives the one which is most economical, or most suitable etc. Good piping system design is always a mixture of sound knowledge base in the basics and a lot of ingenuity. This paper attempts to create the necessary base.

CLASSIFICATION OF LOADS AND FAILURE MODES

Pressure design of piping or equipment uses one criteria for design. Under a steady application of load (e.g. pressure), it ensures against failure of the system as perceived by one of the failure theories. If a pipe designed for a certain pressure experiences a much higher pressure, the pipe would rupture even if such load (pressure) is applied only once. The failure or rupture is sudden and complete. Such a failure is called catastrophic failure. It takes place only when the load exceeds far beyond the load for which design was carried out. Over the years, it has been realized that

systems, especially piping systems can fail even when the loads are always under the limits considered safe, but the load application is cyclic (e.g. high pressure, low pressure, high pressure,..). Such a failure is not guarded against by conventional pressure design formula or compliance with failure theories. Once this was realized and it was seen than systems may fail after prolonged use under the load they could withstand till that time, it became clear that system design must comply with at least two different types of loads causing two different types of failures. For piping system design, it is now well established that one must treat these two types of loads separately and together guard against catastrophic and fatigue failure.

The loads the piping system (or for that matter any structural part) faces are broadly classified as primary loads and secondary loads. There examples and characteristics are given here in brief.

Primary Loads

These are typically steady or sustained types of loads such as internal fluid pressure, external pressure, gravitational forces acting on the pipe such as weight of pipe and fluid, forces due to relief or blow down, pressure waves generated due to water hammer effects. The last two loads are not necessarily sustained loads. All these loads occur because of forces created and acting on the pipe. In fact, primary loads have their origin in some force acting on the pipe causing tension, compression, torsion etc leading to normal and shear stresses. Too large a load of this type leads to deformation, often plastic. The deformation is limited only if the material shows strain hardening characteristics. If it has no strain hardening property or if the load is so excessive that the plastic instability sets in, the system would continue to deform till rupture. One says, that primary loads are not self limiting. It means that the stresses continue to exist as long as the load persists and deformation does not stop because the system has deformed into a no-stress condition but because strain hardening has come into play.

The design to guard against failure by primary loads is based on one or more failure theories such as the ones discussed later in this paper.

Secondary Loads

Just as the primary loads have their origin in some force, secondary loads are caused by displacement of some kind. For example, the pipe connected to a storage tank may be under load if the tank nozzle to which it is connected moves down due to tank settlement. Similarly, pipe connected to a vessel is pulled upwards because the vessel nozzle moves up due to vessel expansion. Also, a pipe may vibrate due to vibrations in the rotating equipment it is attached to. A pipe may experience expansion or contraction once it is subjected to temperatures higher or lower respectively as compared to temperature at which it was assembled.

The secondary loads are often cyclic but not always. For example load due to tank settlement is not cyclic. The load due to vessel nozzle movement during operation is cyclic because the displacement is withdrawn during shut-down and resurfaces again after fresh start-up. A pipe subjected to a cycle of hot and cold fluid similarly undergoes cyclic loads and deformation.

Failure under such loads is often due to fatigue and not catastrophic in nature.

Broadly speaking, catastrophic failure is because individual crystals or grains were subjected to stresses, which the chemistry and the physics of the solid could not withstand. Fatigue failure is often because the grains collectively failed because their collective characteristics (for example entanglement with each other etc.) changed due to cyclic load. Incremental damage done by each cycle to their collective texture accumulated to such levels that the system failed. In other words, catastrophic failure is more at microscopic level, whereas fatigue failure is at mesoscopic level if not at macroscopic level.

PIPING ENGINEERING CELL

This part of the paper focuses more on primary loads and catastrophic failure. A brief implication of cyclic loads and fatigue failure on design is also presented. The subsequent parts would deal more comprehensively with secondary loads including thermal loads and stress analysis concepts.

THE STRESSES

The MoC of any piping system is the most tortured non-living being right from its birth. Leaving the furnace in the molten state, the metal solidifies within seconds. It is a very hurried crystallization process. The crystals could be of various lattice structural patterns such as BCC, FCC, HCP etc. depending on the material and the process. The grains, crystals of the material have no time or chance to orient themselves in any particular fashion. They are thus frozen in all random orientations in the cold harmless pipe or structural member that we see.

When we calculate stresses, we choose a set of orthogonal directions and define the stresses in this co-ordinate system. For example, in a pipe subjected to internal pressure or any other load, the most used choice of co-ordinate system is the one comprising of axial or longitudinal direction (L), circumferencial (or Hoope's) direction (H) and radial direction (R) as shown in Fig.1. Stresses in the pipe wall are expressed as axial (S_L), Hoope's (S_H) and radial (S_R). These stresses, which stretch or compress a grain/crystal, are called normal stresses because they are normal to the surface of the crystal.

But, all grains are not oriented as the grain in Fig.1. In fact the grains would have been oriented in the pipe wall in all possible orientations. The above stresses would also have stress components in direction normal to the faces of such randomly oriented crystal. Each crystal thus does face normal stresses. One of these orientations must be such that it maximizes one of the normal stresses. The mechanics of solids state that it would also be orientation, which minimizes some other normal stress.

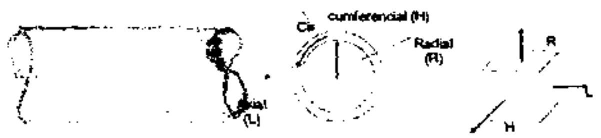

Fig. 1: Commonly Used Coordinate System

The mechanics of solids state that it would also be orientation, which minimizes some other normal stress. Normal stresses for such orientation (maximum normal stress orientation) are called principal stresses, and are designated S_1 (maximum), S_2 and S_3 (minimum). Solid mechanics also states that the sum of the three normal stresses for all orientation is always the same for any given external load. That is

$$S_L + S_H + S_R = S_1 + S_2 + S_3$$

Importance of principal stresses can be stressed at this time. Assume that a material can be deemed to fail of any normal stress exceeds some threshold value. If conventional co-ordinate axes are used, one may find for certain stress state that S_L, S_H and S_R are within this threshold limit. The design would then appear to be safe. However, grains, which are oriented in maximum normal stress orientation, may have one of the stresses (S_1) more than this threshold. The pipe would thus fail as far as these grains are concerned. Design has to be safe for such worst case scenario. Principal stresses are thus a way of defining the worst case scenario as far as the normal stresses are concerned.

In addition to the normal stresses, a grain can be subjected to shear stresses as well. These act parallel to the crystal surfaces as against perpendicular direction applicable for normal stresses. Shear stresses occur if the pipe is subjected to torsion, bending etc. Just as there is an orientation for which normal stresses are maximum, there is an orientation, which maximizes shear stress. The maximum shear

stress in a 3-D state of stress can be shown to be

$$\tau_{max} = (S_1 - S_3)/2$$

i.e. half of the difference between the maximum and minimum principal stresses. The maximum shear stress is important to calculate because failure may occur or may be deemed to occur due to shear stress also. A failure perception may stipulate that maximum shear stress should not cross certain threshold value. It is therefore necessary to take the worst case scenario for shear stresses also as above and ensure against failure.

It is easy to define stresses in the co-ordinate system such as axial-Hoope's-radial (L-H-R) that we define for a pipe. The load bearing cross-section is then well defined and stress components are calculated as ratio of load to load bearing cross-section. Similarly, it is possible to calculate shear stress in a particular plane given the torsional or bending load. What are required for testing failure-safe nature of design are, however, principal stresses and maximum shear stress. These can be calculated from the normal stresses and shear stresses available in any convenient orthogonal co-ordinate system.

In most pipe design cases of interest, the radial component of normal stresses (S_R) is negligible as compared to the other two components (S_H and S_L). The 3-D state of stress thus can be simplified to 2-D state of stress. Use of Mohr's circle then allows to calculate the two principle stresses and maximum shear stress as follows.

$$S_1 = (S_L + S_H)/2 + [\{(S_L - S_H)/2\}^2 + \tau^2]^{0.5}$$
$$S_2 = (S_L + S_H)/2 - [\{(S_L - S_H)/2\}^2 + \tau^2]^{0.5}$$
$$\tau_{max} = 0.5\,[(S_L - S_R)^2 + 4\,\tau^2]^{0.5}$$

$S_3 = 0$ (in piping design)

The third principle stress (minimum i.e. S_3) is zero.

All failure theories state that these principle or maximum shear stresses or some combination of them should be within allowable limits for the MoC under consideration. To check for compliance of the design would then involve relating the applied load to get the net S_H, S_L, τ and then calculate S_1, S_2 and τ_{max} and some combination of them.

NORMAL AND SHEAR STRESSES FROM APPLIED LOAD

As said earlier, a pipe is subjected to all kinds of loads. These need to be identified. Each such load would induce in the pipe wall, normal and shear stresses. These need to be calculated from standard relations. The net normal and shear stresses resulting from all such actual and potential loads are then arrived at and principle and maximum shear stresses calculated. Some potential loads faced by a pipe and their relationships to stresses are summarized here in brief.

Axial Load

A pipe may face an axial force (F_L) as shown in Fig. 2. It could be tensile or compressive.

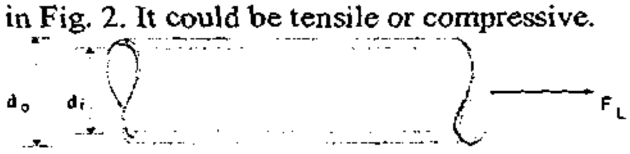

Fig.2: Pipe Under Axial Load

What is shown is a tensile load. It would lead to normal stress in the axial direction (S_L). The load bearing cross-section is the cross-sectional area of the pipe wall normal to the load direction, A_m. The stress can then be calculated as

$$S_L = F_L / A_m$$

The load-bearing cross-section may be calculated rigorously or approximately as follows.

$A_m = \pi (d_o^2 - d_i^2) / 4$ (rigorous)

$= \pi (d_o + d_i)t/2$ (based on average diameter)

$= \pi d_o t$ (based on Outer Diameter)

The axial load may be caused due to several reasons. The simplest case is a tall column. The metal cross-section at the base of the column is under the weight of the column section above it including the weight of other column accessories such as insulation, trays, ladders etc. Another example is that of cold spring. Many times a pipeline is intentionally cut a little short than the end-to-end length required. It is then connected to the end nozzles by forcibly stretching it. The pipe, as assembled, is under axial tension. When the hot fluid starts moving through the pipe, the pipe expands and compressive stresses are generated. The cold tensile stresses are thus nullified. The thermal expansion stresses are thus taken care of through appropriate assembly-time measures.

Internal / External Pressure

A pipe used for transporting fluid would be under internal pressure load. A pipe such as a jacketed pipe core or tubes in a Shell & Tube exchanger etc. may be under net external pressure. Internal or external pressure induces stresses in the axial as well as circumferencial (Hoope's) directions. The pressure also induces stresses in the radial direction, but as argued earlier, these are often neglected.

The internal pressure exerts an axial force equal to pressure times the internal cross-section of pipe.

$F_L = P [\pi d_i^2 / 4]$

This then induces axial stress calculated as earlier. If outer pipe diameter is used for calculating approximate metal cross-section as well as pipe cross-section, the axial stress can often be approximated as follows.

$S_L = P d_o / (4 t)$

The internal pressure also induces stresses in the circumferential direction as shown in Fig. 3.

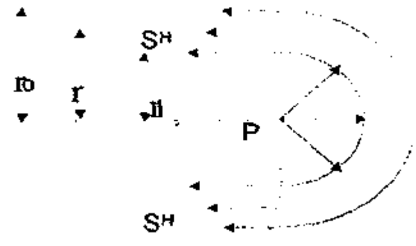

Fig. 3: Hoope's Stress due to Internal Pressure

The stresses are maximum for grains situated at the inner radius and minimum for those situated at the outer radius. The Hoope's stress at any in between radial position (r) is given as follows (Lame's equation)

S_H at $r = P (r_i^2 + r_i^2 r_o^2 / r^2) / (r_o^2 - r_i^2)$

For thin walled pipes, the radial stress variation can be neglected. From membrane theory, S_H may then be approximated as follows.

$S_H = P d_o / 2t$ or $P d_i / 2t$

Radial stresses are also induced due to internal pressure as can be seen from Fig. 4.

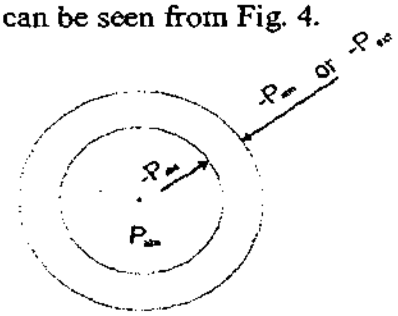

Fig. 4: Radial Stresses Due to Internal Pressure

At the outer skin, the radial stress is compressive and equal to atmospheric pressure (P_{atm}) or external pressure (P_{ext}) on the

(*) never leave a concentrated point without support next to it

pipe. At inner radius, it is also compressive but equal to absolute fluid pressure (P_{abs}). In between, it varies. As mentioned earlier, the radial component is often neglected.

Bending Load

A pipe can face sustained loads causing bending. The bending moment can be related to normal and shear stresses.

Pipe bending is caused mainly due to two reasons: Uniform weight load and concentrated weight load. A pipe span supported at two ends would sag between these supports due to its own weight and the weight of insulation (if any) when not in operation. It may sag due to its weight and weight of hydrostatic test fluid it contains during hydrostatic test. It may sag due to its own weight, insulation weight and the weight of fluid it is carrying during operation. All these weights are distributed uniformly across the unsupported span and lead to maximum bending moment either at the center of the span or at the end points of the span (support location) depending upon the type of the support used.

Let the total weight of the pipe, insulation and fluid be W and the length of the unsupported span be L (see Fig. 5).

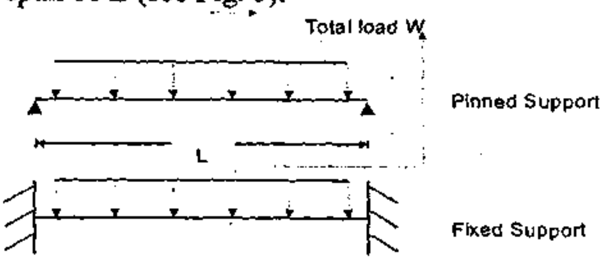

Fig. 5: Distributed Load

The weight per unit length, w, is then calculated ($w = W/L$). The maximum bending moment (M_{max}), which occurs at the center for the pinned support, is then given by the beam theory as follows.

$M_{max} = w L^2/8$ for pinned support

For fixed supports, the maximum bending moment occurs at the ends and is given by beam theory as follows.

$M_{max} = w L^2 / 12$ for fixed support

The pipe configuration and support types used in process industry do not confirm to any of these ideal support types and can be best considered as somewhere in between. As a result, a common practice is to use the following average formula to calculate bending moment for practical pipe configurations as follows.

$M_{max} = w L^2 / 10$

Also, the maximum bending moment in the case of actual supports would occur somewhere between the ends and the middle of the span.

Another load that the pipe span would face is the concentrated load. A good example is a valve on a pipe run (see Figure 6).

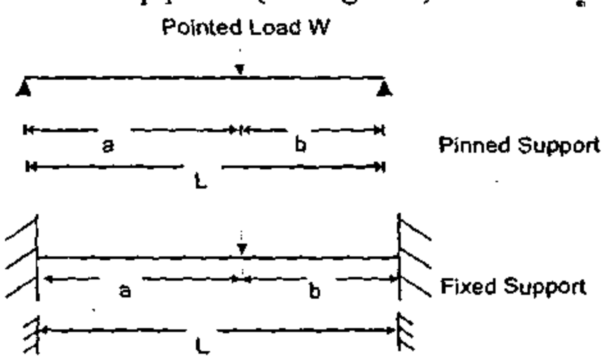

Fig. 6: Pointed Load

The load is then approximated as acting at the center of gravity of the valve and the maximum bending moment occurs at the point of loading for pinned supports and is given as

$M_{max} = W a b / L$

For rigid supports, the maximum bending moment occurs at the end nearer to the pointed load and is given as

(larger [illegible] = a [illegible] b)

$M_{max} = W a^2 b / L$

A is to be taken as the longer of the two arms (a and b) in using the above formula.

As can be seen, the bending moment can be reduced to zero by making either a or b zero, i.e. by locating one of the supports right at the point where the load is acting. In actual practice, it would mean supporting the valve itself. As that is difficult, it is a common practice to locate one support as close to the valve (or any other pointed and significant load). With that done, the bending moment due to pointed load is minimal and can be neglected.

Whenever the pipe bends, the skin of the pipe wall experiences both tensile and compressive stresses in the axial direction as shown in Fig. 7.

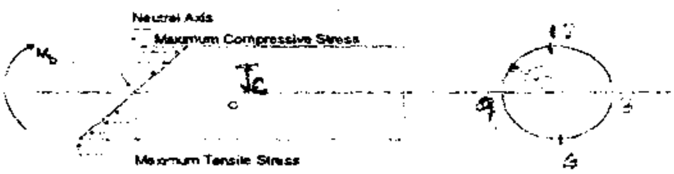

Fig. 7: Axial stresses due to Bending

The axial stress changes from maximum tensile on one side of the pipe to maximum compressive on the other side. Obviously, there is a neutral axis along which the bending moment does not induce any axial stresses. This is also the axis of the pipe.

The axial tensile stress for a bending moment of M_b at any location c as measured from the neutral axis is given as follows.

$S_L = M_b c / I$

I is the moment of inertia of the pipe cross-section. For a circular cross-section pipe, I is given as

$I = \pi(d_o^4 - d_i^4)/64$

The maximum tensile stress occurs where c is equal to the outer radius of the pipe and is given as follows.

S_L at outer radius $= M_b r_o / I = M_b / Z$

where $Z (= I/r_o)$ is the section modulus of the pipe.

Shear Load

Shear load causes shear stresses. Shear load may be of different types. One common load is the shear force (V) acting on the cross-section of the pipe as shown in Fig. 8.

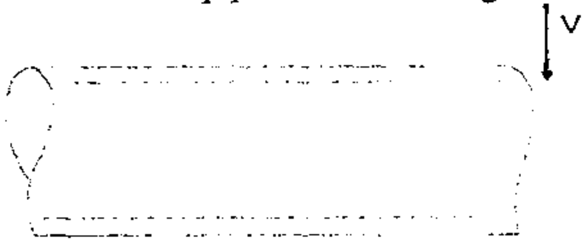

Fig. 8: Shear Force on a Pipe

It causes shear stresses, which are maximum along the pipe axis and minimum along the outer skin of the pipe. This being exactly opposite of the axial stress pattern caused by bending moment and also because these stresses are small in magnitude, these are often not taken in account in pipe stress analysis. If necessary, these are calculated as

$\tau_{max} = V Q / A_m$

where Q is the shear form factor and A_m is the metal cross-section.

Torsional Load

This load (see Fig. 9) also causes shear stresses. The shear stress caused due to torsion is maximum at outer pipe radius. And is given there in terms of the torsional moment and pipe dimensions as follows.

two ... at ... diff. elevation
if one at higher elevation expands, it causes the other end to come under tension)

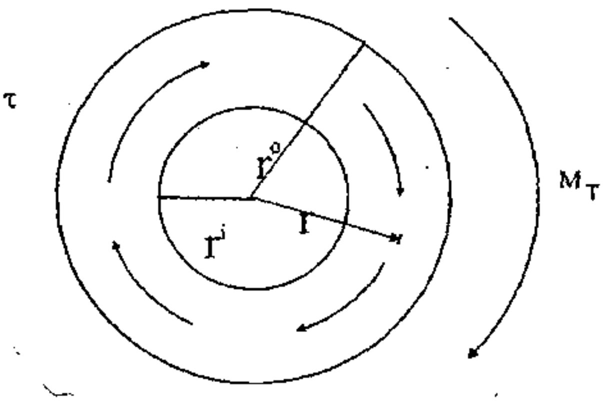

Fig. 9: Shear Force Due to Torsion

$\tau\,(\text{at } r = r_o) = M_T\, r_o / R_T = M_T\, r_o / (2I) = M_T / 2Z$

R_T is the torsional resistance (= twice the moment of inertia).

All known loads on the pipe should be used to calculate contributions to S_L, S_H and τ. These then are used to calculate the principal stresses and maximum shear stress. These derived quantities are then used to check whether the pipe system design is adequate based on one or more theories of failure.

THEORIES OF FAILURE

A piping system in particular or a structural part in general is deemed to fail when a stipulated function of various stresses and strains in the system or structural part crosses a certain threshold value. It is a normal practice to define failure as occurring when this function in the actual system crosses the value of a similar function in a solid rod specimen at the point of yield. There are various theories of failure that have been put forth. These theories differ only in the way the above-mentioned function is defined. Important theories in common use are considered here.

Maximum Stress Theory

This is also called Rankine Theory. According to this theory, failure occurs when the maximum principle stress in a system (S_1) is greater than the maximum tensile principle stress at yield in a specimen subjected to uniaxial tension test.

Uniaxial tension test is the most common test carried out for any MoC. The tensile stress in a constant cross-section specimen at yield is what is reported as yield stress (S_Y) for any material and is normally available. In uniaxial test, the applied load gives rise only to axial stress (S_L) and S_H and S_R as well as shear stresses are absent. S_L is thus also the principle normal stress (i.e. S_1). That is, in a specimen under uniaxial tension test, at yield, the following holds.

$S_L = S_Y$, $S_H = 0$, $S_R = 0$
$S_1 = S_Y$, $S_2 = 0$ and $S_3 = 0$

$\tau = 0$)

The maximum tensile principle stress at yield is thus equal to the conventionally reported yield stress (load at yield / cross-sectional area of specimen).

The Rankine theory thus just says that failure occurs when the maximum principle stress in a system (S_1) is more than the yield stress of the material (S_Y).

The maximum principle stress in the system should be calculated as earlier.

It is interesting to check the implication of this theory on the case when a cylinder (or pipe) is subjected to internal pressure.

As per the membrane theory for pressure design of cylinder, as long as the Hoope's stress is less that the yield stress of the MoC, the design is safe. It is also known that Hoope's stress (S_H) induced by internal

Rankine Theory

pressure is twice the axial stress (S_L). The principle stresses in the cylinder as per the earlier given formula would be

$$S_1 = (S_L + S_H)/2 + [\{(S_L - S_H)/2\}^2 + \tau^2]^{0.5}$$
$$= S_L \quad (2S_L)$$
$$S_2 = (S_L + S_H)/2 - [\{(S_L - S_H)/2\}^2 + \tau^2]^{0.5}$$
$$= S_H$$

The maximum principle stress in this case is S_2 ($=S_H$). The Rankine theory and the design criterion used in the membrane theory are thus compatible.

Check that the same is the case if we consider the design formula for sphere based on membrane theory. Membrane theory widely used for pressure thickness calculation for pressure vessels and piping design uses Rankine theory as a criterion for failure.

Maximum Shear Theory

This is also called Tresca theory. According to this theory, failure occurs when the maximum shear stress in a system (Tmax) is greater than the maximum shear stress at yield in a specimen subjected to uni-axial tension test. Note that it is similar in wording to the statement of the earlier theory except that maximum shear stress is used as criterion for comparison as against maximum principle stress used in the Rankine theory.

In uniaxial test, the maximum shear stress at yield as per definition of maximum shear test given earlier is

$$\tau_{max} = 0.5\,[(S_L - S_H)^2 + 4\,\tau^2]^{0.5}$$
$$= S_L/2 = S_Y/2$$

The Tresca theory thus just says that failure occurs when the maximum shear stress in a system (τ_{max}) is more than half the yield stress of the material (S_Y).

The maximum shear stress in the system should be calculated as earlier.

It should also be interesting to check the implication of this theory on the case when a cylinder (or pipe) is subjected to internal pressure.

As the Hoope's stress induced by internal pressure (S_H) is twice the axial stress (S_L) and the shear stress is not induced directly ($\tau = 0$) the maximum shear stress in the cylinder as per the earlier given formula would be

$$\tau_{max} = 0.5\,[(S_L - S_H)^2 + 4\,\tau^2]^{0.5}$$
$$= 0.5\,S_H \quad (0.5\,S_L)$$

This should be less than 0.5 S_Y as per Tresca theory for safe design. This leads to the same criterion that Hoope's stress in a cylinder should be less than yield stress. The Tresca theory and the design criterion used in the membrane theory for cylinder are thus compatible.

Check whether the same is the case if we consider the design formula for sphere based on membrane theory.

Octahedral Shear Theory

This is also called von Mises theory. According to this theory, failure occurs when the octahedral shear stress in a system (τ_{oc}) is greater than the octahedral shear stress at yield in a specimen subjected to uniaxial tension test. Note that it is similar in wording to the statement of the earlier two theories except that octahedral shear stress is used as criterion for comparison as against maximum principle stress used in the Rankine theory or maximum shear stress used in Tresca theory.

The octahedral shear stress is defined in terms of the three principle stresses as follows.

$$\tau_{oct} = 1/3\,[(S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2]^{0.5}$$

(more of civil engineering theory)

In view of the principle stresses defined for a specimen under uniaxial load earlier, the octahedral shear stress at yield in the specimen can be shown to be as follows.

$$\tau_{oct} = 2^{0.5} S_Y / 3$$

The von Mises theory thus states that failure occurs in a system when octahedral shear stress in the system exceeds $2^{0.5} S_Y / 3$.

The reader should check what it implies for the case of cylinder and sphere and how it compares with the membrane theory criteria for design.

For stress analysis related calculations, most of the present day piping codes uses a modified version of Tresca theory.

DESIGN UNDER SECONDARY LOAD

As pointed earlier, a pipe designed to withstand primary loads and to avoid catastrophic failure may fail after a sufficient amount of time due to secondary cyclic load causing fatigue failure. The secondary loads are often cyclic in nature. The number of cycles to failure is a property of the material of construction just as yield stress is. While yield stress is cardinal to the design under primary sustained loads, this number of cycles to failure is the corresponding material property important in design under cyclic loads aimed at ensuring that the failure does not take place within a certain period for which the system is to be designed.

While yield stress is measured by subjecting a specimen to uniaxial tensile load, fatigue test is carried out on a similar specimen subjected to cycles of uniaxial tensile and compressive loads of certain amplitude, i.e. magnitude of the tensile and compressive loads. Normally the tests are carried out with zero mean load. This means, that the specimen is subjected to a gradually increasing load leading to a maximum tensile load of W, then the load is removed gradually till it passes through zero and becomes gradually a compressive load of W (i.e. a load of -W), then a tensile load of W and so on. Time averaged load is thus zero. The cycles to failure are then measured. The experiments are repeated with different amplitudes of load. The results would be typically as in Table 1.

Table 1: Typical Fatigue Test Results

Experiment Number	Applied Cyclic Stress, psi	Cycles to Failure
1	20000	100,000
2	30000	38000
3	50000	6700
4	100000	530
5	200000	90
6	300000	23

This table was for a MoC with yield stress of 57000 psi. Some interesting observations can be made and questions raised.

If the material has a reported yield stress of 57000, how were stresses far more than that number created during fatigue tests on the specimen as reported in the above table? This question is very common and natural for all those who do stress analysis and observe reported stresses at various nodes of a piping system which are often far beyond the yield stress. In the stress-strain curve generated for the specimen using uniaxial tensile load, such a possibility would not be seen because for any stress more than the yield stress, the material would seem to strain more and more without allowing a possibility of significantly increasing the stress further. This question can be answered as follows.

One must always remember that stresses are always derived rather than actually measured quantities. What is actually measured is the load or the strain. The stresses are either reported as applied load divided by the original load-bearing cross-section or the values corresponding to the observed strain as noted on the elastic line's intersection with the strain vertical. What the later gives is the hypothetical stress that would have been

generated had the material stayed in the elastic region and still produced that much strain. The stress calculated in this way is called code stress and what are reported are code stresses. In actual practice, material would crossover to the plastic deformation range and cause observed strain for much smaller actual stress.

Another observation is that even when the loads on the piping system are far below the yield stress (say 30000 psi), the system would fail after a certain number of cycles. The design approach based on primary loads and guarding against catastrophic failure is thus simply not adequate for cyclic load.

When the amplitude of the cyclic stress is approximately the same as the yield stress for the material, the number of cycles to failure is about 7000. What does that mean in terms of real time? What would be the life of a piping system or its component, which is subjected to such a stress cycle? That would depend on the frequency or period of the stress cycle. That in turn would depend on the process and operating philosophy. If the process is such that the cycle period is 24 hours and the process operates round the year, then there are 365 cycles per year. The cycles would cross 7000 in about 20 years and then the system would fail due to fatigue. This means that if we have an operation, which requires start up, and shut down every day, the life of the plant designed for sustained load would be about 20 years. For processes, which have larger periodicity of stress cycles, the fatigue life would be proportionately smaller. For example, if the process requires a shut down and start up in every 8 hour shift and the plant operates three shifts a day and 365 days a year, The fatigue life of a component subject to cyclic load due to this cyclic operation would be just 7 years or so if the component is stressed to yield stress in each such cycle. If the life still has to be 20 years, the component must be designed for smaller stress level (i.e. of larger thickness) so that it requires more cycles to failure (approximately 21000). If the allowable stress taken in design for sustained load is appropriately reduced, desired fatigue life also can be achieved. The cyclic stress vs. cycles to failure data is thus useful to decide the factor by which allowable stress should be reduced to guard any design against catastrophic as well as fatigue failure. This consideration is behind the cyclic reduction factor associated with the stress analysis.

It has been shown that cycles to failure are also a function of the mean stress. For example, a particular system may be cycling between 50000 psi to 20000 psi stress, both tensile. The mean stress is thus 35000 psi and not zero. Under this sustained mean stress, fatigue failure would take place much earlier than under zero mean stress. Laboratory tests are also carried out to study the effect of sustained load over and above which the cyclic load is imposed.

CONCLUSION

Stresses in pipe or piping systems are generated due to loads experienced by the system. These loads can have origin in process requirement; the way pipes are supported, piping system's static properties such as own weight or simple transmitted loads due to problems in connecting equipments such as settlement or vibrations. Whatever may be the origin of load, these stress the fabric of the MoC and failure may occur. This paper attempted to present a rather simplistic view of common loads and their implications on stresses and failure.

Fatigue failure is an important aspect in flexibility analysis of piping systems. Often cyclic stresses in piping systems subjected to thermal cycles get transferred to flexibility providing components such as elbows. These become the components susceptible to fatigue failure. Thermal stress analysis or flexibility analysis attempts to guard against such failure through very involved calculations. That is the subject matter of a series of papers to appear in the columns of this journal.

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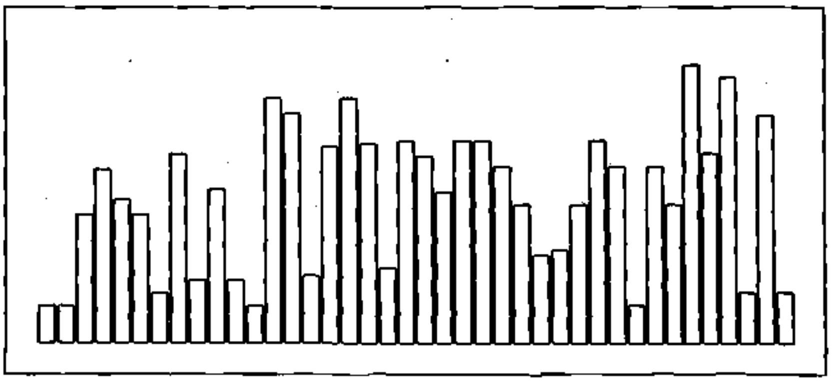
Certificate Course
on
PIPING ENGINEERING

June 12 - 25, 2006

STRESS ANALYSIS

T. N. Gopinath
Consultant



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STRESS ANALYSIS

T.N.GOPINATH

1.0 INTRODUCTION

Stress Analysis is a subject, which is more talked about and less understood. The objective of pipe stress analysis is to ensure safety against failure of the **Piping System** by verifying the structural integrity against the loading conditions, both external and internal, expected to occur during the lifetime of the system in the plant. This is to be undertaken with the most economic considerations. Hence the objectives of stress Analysis could be listed as

1.1 Objectives of stress Analysis are to

1.1.1 Ensure that the stresses in the piping components in the system are within the allowable limits.

1.1.2 Solve dynamic problems developed due to mechanical vibration, acoustic vibration, fluid hammer, pulsation, relief valves etc.

1.1.3 Solve the problems associated due to higher or lower operating temperature such as

a) Displacement **Stress range**

b) Nozzle loading on the connected equipment

c) Pipe displacements

d) Loads and moments on the supporting structures

When piping is connected to strain sensitive equipment, the flexibility required to satisfy the acceptable limits of nozzle loading on the connected equipment ('b' above) overrides all other considerations.

1.2 Classifications of piping systems

The piping systems are mainly classified into three main categories and then again subcategories. The main categories are the

1.2.1 Hot Systems.

1.2.2 Cold Systems.

1.2.3 Cryogenic Systems.

The fundamental reason for this classification is that hot lines and cryogenic lines must undergo ***Flexibility analysis*** to determine thermal forces, displacements and stresses. These systems are further divided into. .

i) Small bore lines

ii) Large (Big) bore lines.

As a general practice those pipe lines with nominal diameters 40mm (1½") NB and under are classified as small and 50mm (2") NB and above as large. Further, piping systems could be classified based on the regulatory codes under which the system is designed. Certain codes require more stringent analysis than others.

1.3 Hence the steps involved in the stress analysis can be listed as.

1.3.1. Identify the potential loads that the piping system would encounter during the life of the plant.

1.3.2. Relate each of these loads to the stresses and strains developed.

1.3.3. Get the cumulative effect of the potential loads in the system.

1.3.4. Decide the allowable limits, the system can withstand without failure.

1.3.5. After the system is designed, to ensure that the stresses are within the safe limits.

1.4 Types of loads

All the American code for Pressure Piping classify the loads mainly into three types.

1.4.1. Sustained Loads: Those due to forces present during normal operation (Longitudinal, hoop's, Bending, Shear)

1.4.2. Occasional Loads: Those present during rare intervals of operations

1.4.3. Displacement Loads: Those due to displacement of pipe

These are dealt with in detail in the chapter **Pipe Under stress.** Hence the content of this chapter is limited to the details of analysis of piping system under the sustained and displacement loads. This analysis is most commonly called as the **Flexibility Analysis.** Further those conditions stipulated in the regulatory code ASME B 31.1 and ASME B 31.3 only are considered hereafter.

1.5 Conditions of Acceptability of Piping System

The Piping Engineer has the following choices to establish that the required flexibility has been provided in the piping layout.

1.5.1 As per clause 119.7.1/319.4.1 of the code ASME B 31.1/B 31.3, no formal analysis is required in systems which

i) Are duplicates of successfully operating installations or replacements.

ii) Can readily be judged adequate by comparison with previously analyzed systems.

iii) Satisfy equation specified in clause 119.7.1(A3)/ 319.4.1 (c) .

1.5.2 Analyzing the layout by an approximate method.

Approximate method shall be applied only if they are used for the range of configuration for which adequate accuracy has been demonstrated.

1.5.3 Carrying out a comprehensive analysis.

i) Analytical

ii) Model test

iii) Chart method

2.0 CODE COMPLIANCE

Let us consider those aspects in the code, which are mandatory requirements for the expansion and flexibility of metallic piping. The Piping Specification nominates the code to be used for various aspects in the Piping System. Let us consider those, which are of importance to the Piping Engineer to carry out the flexibility analysis. Every such code will contain recommendations and mandatory requirements on the following aspects.

i) Minimum flexibility requirements for thermal expansion

ii) Allowable stresses for various piping materials

iii) Reinforcement requirements of branch connections

iv) Support criteria

It is the responsibility of the Piping Engineer to demonstrate the compliance with these

wind load, seismic load, etc are occasional load

requirements and achieve the most economical, safe and practical layout.

Let us consider each of the above in turn to see how they are dealt with by a Piping Engineer.

2.1 Installed And Operating Temperature

Pipe is erected at ambient temperature and that depends on the climatic conditions. 70°F (21°C) is the figure commonly used for calculations. The same piping when in operation in a Petrochemical Plant could achieve a temperature in excess of 500°C if it were in a reactor piping system or it could be of the order of -120°C, if it were associated with ethylene refrigeration system.

2.1.1 Displacement stresses.

A piping system will undergo dimensional changes with any change in temperature. If it is constrained from free expansion or contraction, it will be displaced from it unrestrained position causing strain and stresses. The system could behave either balanced or unbalanced under such conditions.

a) Balanced System

Displacement strains are well distributed and not excessive at any point. Layout of the system should aim at such condition, which is assumed in flexibility analysis methods provided in the code.

b) Unbalanced System

In an unbalanced system, stress cannot be considered proportional to displacement strains through out a piping system in which an excessive amount of strain may occur in a localized portion of the system. Unbalance may result from

i) Highly stressed small pipe run in series with large or relatively stiff pipe runs.

ii) Local reduction in size or wall thickness.

iii) Line configuration in a system of uniform size in which the expansion or contraction must be absorbed largely in short offset.

Unbalance must be avoided by design and layout of piping system. If unbalance cannot be avoided the designer shall use appropriate analytical methods as specified in the code to assure adequate flexibility.

Each material has its own coefficient of thermal expansion. These values are given in Appendix C of the code ASME B 31.3 and Appendix B of the code ASME B 31.1. If the pipe is of carbon steel or of low alloy steel, it will expand at the rate of 6 to 7 mm every meter length as the temperature rises to 500^{0}C. This means that the pipe running between two equipments 10m apart may well want to expand by 60 to 70 mm or more as it heats up. The increased length can be accommodated only by straining the pipe as the ends are not free to move. This straining, if not freely allowed, induces stresses in the pipe as well as load at the support points. However, when the line is cooled during shutdown to ambient temperature the expansion returns to zero, the straining is no longer required and hence the load and stress also disappears. This can be demonstrated as below.

2.2 Magnitude Of Thermal Load

A pipe line (Fig. 2.1), held between two anchors, when heated up, tries to expand against its restraints resulting in forces, moments and stresses.

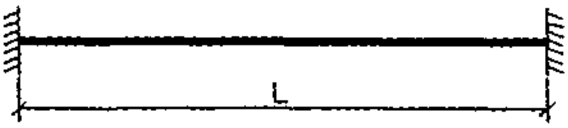


Fig. 2.1

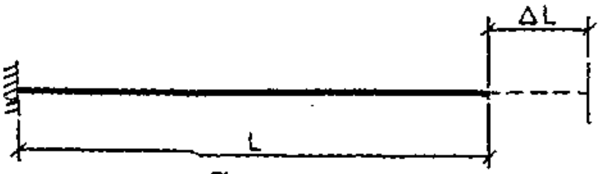


Fig. 2.2

If the pipe is allowed to expand freely due to rise in its temperature, it would expand by ΔL as shown in Fig. 2.2. . The free expansion will take place when one of the anchors is released. ΔL would depend upon the pipe length (L), temperature rise (difference between the temperature under hot condition and initial cold condition and also the length of the pipe). ΔL can be calculated if the coefficient of thermal expansion, an important physical property of any material, is known. The axial force generated in the above configuration can be estimated to be the axial force required to compress the pipe back to its original position from its expanded position.

In the textbooks, coefficient of linear thermal expansion is defined as the increase in length of a specimen of unit length, if it is subjected to a temperature rise of 1^0C and is often designated as α. If a pipe of length L is heated to a temperature which is ΔT^0C above its installed temperature, the increase in length would then be

$$\Delta L = \alpha \cdot \Delta T \cdot L \quad \ldots \quad (1)$$

In the codes and many reported calculations, however, α is used as inclusive of ΔT (i.e. $\alpha.\Delta T$ in above equation) and is called coefficient of thermal expansion from installed/ambient to operating temperature.

(Coeff. of expansion = $\alpha \Delta T$)

The expansion is then written as

$$\Delta L = \alpha \cdot L \quad \ldots \quad (2)$$

This difference in meaning of α used in equations 1 and 2 should be taken note of. A typical Table C-1 of total thermal expansion from ASME 31.3 for a specific material is as follows. It can be used to get α easily for the applicable difference in operating and installed temperature.

Table C-1ASME B 31.3

TOTAL THERMAL EXPANSION, US UNITS, FOR METALS

Total Linear Thermal Expansion Between 70^0F and Indicated Temperature, in./100 ft

	Material				
Temp., ^{0}F	Carbon Steel...	Austenitic Stainless Steel...	—	—	Copper and Copper Alloys
25	-0.32	-0.46	--	--	-0.50
50	-0.14	-0.21	--	--	-0.22
70	0	0	--	--	0
100	0.23	0.34	--	--	0.34
125	0.42	0.62	--	--	0.63

If the pipe is to be maintained in the original position then there will be an axial force P to compress the increase in pipe length of ΔL (Fig. 2.3).

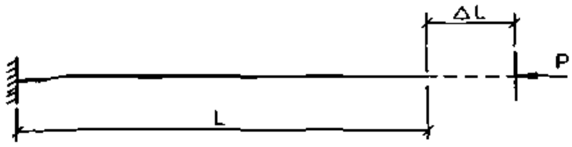


Fig. 2.3

The strain developed in the pipe, ε, is then calculated as $\varepsilon = \Delta L / L = \alpha$

Internal stress developed due to this strain,
$f = E\varepsilon$ (Hooke's Law) $= E\alpha$
The force required to compress back is
$P = Af = AE\alpha$

where A = Area of cross section of pipe, In^2 (mm^2)

E = Modulus of elasticity of material, psi (Kpa)
P = Compressive force on pipe, lbs (N)
f = Stress developed, psi (Kpa)
ΔL = Axial compression of pipe, In (mm)
L = length of pipe, In (mm)

To evaluate the magnitude of such a force, let us consider Carbon Steel pipe of 600 mm outside diameter with 10 mm thickness, operating at a temperature of 300°C.

Referring to ASME B 31.3, Table C6,
$E = 26.85$ Msi (1.888×10^4 kg/mm^2)

Referring to Table C1
$\alpha = 3.625 \times 10^{-3}$ mm/mm

Area of the pipe
$A = \text{Pi} / 4\,[(600)^2 - (580)^2]$
$= 18535.4$ mm^2

$P = 18535.4 \times 1.888 \times 10^4 \times 3.625 \times 10^{-3}$
$= 12,68,563$ kg
$= 1269$ tons

As per most of the design codes, there is no stress in the above configuration, since no bending moment is produced in the axial run. However, the possibility of buckling due to the development of compressive stresses must be considered. In no case, such a design could be accepted.

Alternative to this is the piping system with no fixed points, allowing the pipe to expand. However, this is also not feasible, as the equipment cannot float in space. Equipment on wheels is one possibility but seldom practiced. So every time the plant is in operation and during shutdown, the same cycle of events occurs. The pipe starts from stress free condition when cold, gets stressed with stress reaching maximum at operating condition and then the stresses get reduced to zero when operation stops and system cools down.

This was an oversimplified picture and this is not what exactly happens in practical situations. The piping system can absorb large displacement without returning exactly to previous configuration. Relaxation to the sustaining level of material will tend to establish a condition of stability in few cycles, each cycle lowering the upper limit of hot stress until a state of equilibrium is reached in which the system is completely relaxed and capable of maintaining constant level of stress. The stress at which the material is relieved due to relaxation appears as stress in the opposite temperature state, with equal intensity but of opposite sign. Thus the system, which originally was stress less, could within a few cycles accumulate stress in the cold condition and spring itself without the application of external load. This phenomenon is called "Self springing". This can be demonstrated as follows.

Ksi = 10^3 psi
msi = 10^6 psi

If no installation temp is given, Take - 70°F (according to B31.3)

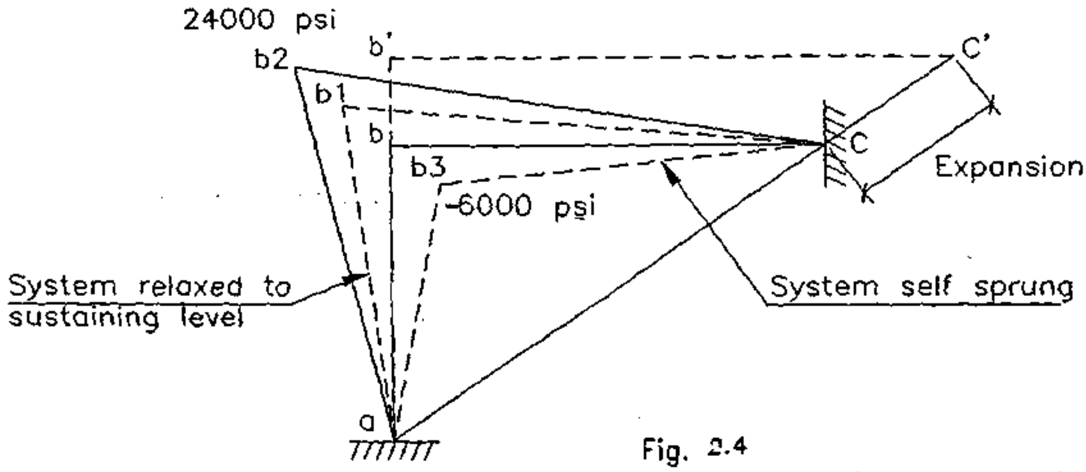


Fig. 2.4

Consider the piping system abc (as installed) as shown in Fig. 2.4. As it is taken from installed condition to hot operating condition, let us assume that the leg 'ab' would expand by 2" and leg 'bc' by 2". This would happen if anchor at 'c' end of the pipe was movable. The pipe would then be in position ab'c' with the anchor c having moved to c' as shown. If both the anchors (at a and c) are rigid, the pipe may still expand and attain position ab_2c. It may then be compressed to bring b_2 back to position b. Such free expansion is however not allowed and let us assume that the system absorbs this 4" expansion between anchors at a and c and the resultant calculated maximum stress is 24,000 psi. Supposing the material at the particular operating temperature can sustain only 18,000 psi or ¾ of this developed stress, yielding will take place and the pipe would be at its sustaining level indicated by ab_1c. We would say that the pipe has absorbed stress of 6000 psi by yielding somewhat. The stresses in the pipe at this stage are only 18000 psi. On cooling back to ambient temperature the system must contract by 4". Contraction would relieve the compressive stresses, which were developed because the pipe wanted to expand by 4" but was not allowed to do so completely. At ¾ of this contraction, i.e., at a net contraction of 3", the system will become stressless. Completion of contraction through remaining 1" will result in a stress of 6000 psi in the opposite direction (tensile stress in this case). The system would now be in position ab_3c as shown in Fig. 2.4. The system, which was stressless at the start at cold condition, will now have residual stresses under cold condition and is said to cold spring. For example, if the pipe was cut anywhere along its route under this condition, it would separate into two segments with violent spring action.

The true magnitude of the stress either in hot or cold condition cannot be determined by simple calculations because the amount of relaxation is unknown and cannot be judged reliably. It depends in a complex way on the metallurgy, pipe route, anchor locations and geometry etc. However, *service failures are related to cyclic rather than static conditions and it is therefore permissible to assume that the system will operate satisfactorily if sum of hot and cold stresses is within a stress range, which is considered safe for an expected number of stress reversals.* This concept provides a logical basis to the design of a piping system because it takes into consideration all the stress levels to which the system is subjected. The actual stress intensity at a specific stage is of academic interest only.

It could be seen that the code recognizes the fact that stresses in piping

system are not necessarily of constant intensity, that the expansion stress at elevated temperature may not be sustained because of relaxation or creep and consequently will drop to stress level the material can sustain. The phenomenon of yielding in the elastic range or flow in the plastic stage presents a problem different from that encountered in the analysis of structure which operates at relatively low temperatures and therefore in a state of steady stress. In this case, the elements are designed to meet the limiting stress or deformation within the elastic limit of the material. In contrast, with stresses from sustained loads such as internal pressure or weight the displacement stresses can cross the elastic limit with stress reversal from cold to hot condition and still remain safe provided the number of stress reversals remains below the limit to exclude the possibility of failure due to fatigue.

While stresses resulting from displacement strains diminish with time due to yielding or creep, the algebraic difference between strains in the extreme displacement condition and as-installed condition remains substantially constant during any one cycle of operation. The difference in strain produces a corresponding stress differential, the **displacement stress range**, which is used as the criteria for designing of piping flexibility.

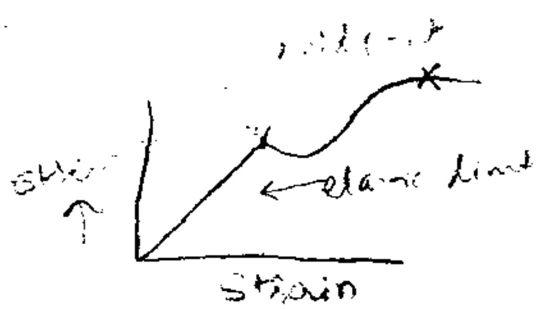
The type of cyclic straining described above, if repeated often enough, will cause the pipe to crack. The cracking will start at a point or points where the stress is maximum. This is what is meant by "Fatigue Failure".

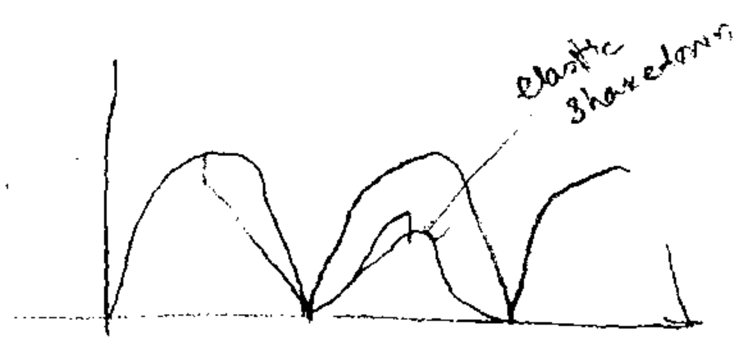
Average axial stress (over the pipe cross section) due to longitudinal forces caused by displacement strains are not normally considered in the determination of displacement stress range since these stresses are not significant in typical piping layouts. In special cases consideration of average axial displacement stress is necessary. Example include buried lines containing hot fluids, double wall pipes and parallel lines with *different* operating temperatures connected together at more than one point.

2.3 Effect of Fatigue on Piping

ARC Markl investigated the phenomenon of fatigue failure of piping during 1940's and 1950's. He tested a number of configurations; straight pipe, various fittings such as elbow, miter bend, welding tee, fabricated tee etc. mostly on 4" NB size by using cyclic displacements to apply alternate bending stress. Plotting the cycles to failure for each applied displacement, he found that the results were on the expected lines and followed the shape of fatigue curves.

Markyl observed that the fatigue failure occurred not in the middle of his test spans, but in the vicinity of fittings and also at lower stress / cycle combinations than for the straight pipes. This lead to what is called the "Stress Intensification Factor" which covered under section 5.3.





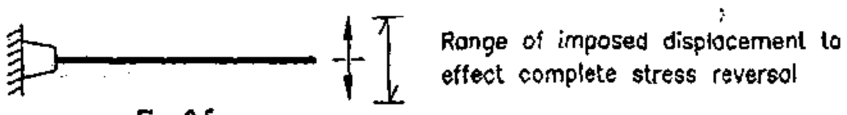


Fig. 2.5

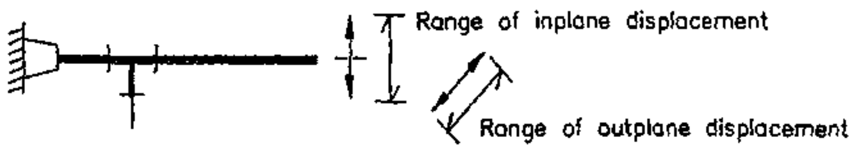


Fig. 2.6

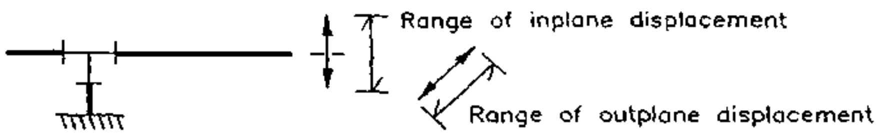


Fig. 2.7

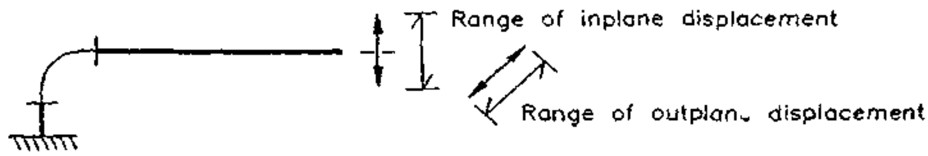


Fig. 2.8

If an initially applied displacement load causes the pipe to yield, it results in plastic deformation, producing prestress in the system when it is brought back to its original state by withdrawing the load. This prestress must be overcome by subsequent stress applications, resulting in lower absolute stresses during the later load cycles. Because of this expected system relaxation, the initial thermal stresses are allowed to exceed the material yield stress, with the aim that the system self springs during the first few cycles and then settles down into elastic cycling. The reason for this allowable over stress is that a repeatedly applied load which initially forces the pipe into the plastic range will, after a few cycles, "shake down" and be reduced to elastic action in the piping system. This theory can be understood by considering a pipe experiencing an imposed displacement which is beyond it's yield strain. When cyclic displacement is removed, the piping component, which exceeded the yield point, will retain the residual distortion equal to imposed strain less yield strain . This distortion will induce yield stress, opposite in sign to that developed during loading and equal to difference between the calculated stress value and the material yield stress (Eε - Sy). The elastic, and therefore the allowable range has been extended by the value of this prestress.

2.4 Allowable Stresses

The American piping codes covered under ASME B 31 subscribe to the failure of the piping system to the basis the **'Maximum principal stress theory'**. The theory states that the yielding occurs when the magnitude of any of the three mutually perpendicular stresses exceeds the yield strength of the material. Temperature and pressure are the significant factors governing the stresses created in the piping systems. There are other factors that influence the stress as well. They are

i) Wind load
ii) Seismic load

SIF - Stress intensity Factor

iii) Relief valve forces
iv) Fluid hammer
v) Settlement
vi) Equipment vibration
vii) Weight of attachments
viii) Weight of contents

All these factors contribute to two distinct forms of stresses.

The sustained stresses – Generated by Pressure, dead weight of contents and attachments, which can be expected to be present virtually at all time of plant operation.
The self-limiting stresses – Generated by thermal effects.

The allowable stresses for these two influences are based on different concepts. However, the allowable stresses specified in code are based on the material properties. They can be classified in two categories as below.

2.4.1 TIME INDEPENDENT STRESSES

Time independent allowable stress is based on either yield stress or the ultimate tensile strength measured in a simple tensile test.

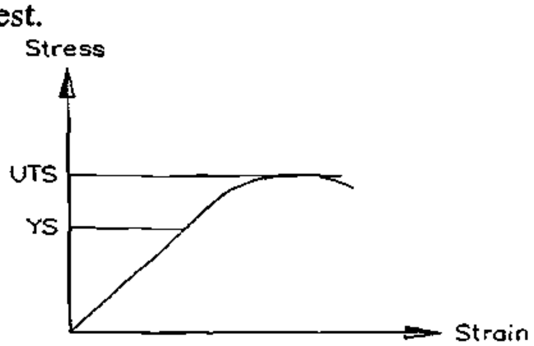


Fig 2.9

The yield stress is the elastic limit and that is the value below which the stresses are proportional to strain and when the load is removed, there is no permanent distortion.

The ultimate tensile strength is the highest stress which the specimen can accommodate without failure.

The basic allowable material stress at the hot (operating/design) temperature (Sh) is defined by the code as minimum of

As per the ASME B 31.1

i) 1/4 of the ultimate tensile strength of the material at operating temperature
ii) 1/4 of the ultimate tensile strength of the material at room temperature
iii) 5/8 of the yield strength of the material at operating temperature (90% of the yield stress for austenitic stainless steels)
iv) 5/8 of the yield strength of the material at room temperature (90% of the yield stress for austenitic stainless steel) and
v) 100% of the average stress for a 0.01% creep rate per 1000 hrs.

As per ASME B 31.3

i) 1/3 of the ultimate tensile strength of the material at operating temperature.
ii) 1/3 of the ultimate tensile strength of the material at room temperature.
iii) 2/3 of the yield strength of the material at operating temperature (90% of the yield stress for austenitic stainless steel)
iv) 2/3 of the yield strength of material at room temperature (90 % of the yield stress for austenitic stainless steel)
v) 100% of the average stress for a 0.01% creep rate at 1000 hrs
vi) 67% of the average stress for rupture after 1,00,000 hrs
vii) 80% of the minimum stress for rupture after 1,00,000 hrs.

Buckling Failure

temp ↑ stress ↓

2.4.2 TIME DEPENDENT STRESSES

Time dependent allowable stress is usually related to the "creep rupture strength" at high temperature. At temperature above 1/3 of the melting point, most metals will exhibit creep in standard tensile test, if the load is kept constant the specimen will continue to deform with time. Under constant load, the rate of creep strain will decrease initially to a steady state and later will increase rapidly until it fails due to creep rupture.

The code uses an allowable stress, which is the smaller of time dependent, and time independent allowable stress. The time dependent allowable stress is the smallest of 67% of the average stress to cause creep rupture in 1,00,000 hrs, 80% of the minimum stress to cause rupture in 1,00,000 hrs or 100% of the stress to give 0.01% of creep rate per hour (Ref. 2.4.1)

The self limiting stress in piping system are essentially cyclic and the initial hot stresses, if they are of sufficient magnitude, will decrease with time because of the plastic strains and will reappear as a stress of reverse direction when the pipe cools. This phenomenon forms the basic difference between the self-limiting stresses and the sustained stresses. (see fig - 2.4)

The degree of self-springing, as explained earlier, will depend on the magnitude of the initial hot stresses and the temperature, so that while the hot stresses will gradually decrease with time, the sum of hot and cold stresses will stay the same. This sum is called the **EXPANSION STRESS RANGE**. This leads us to the selection of an **ALLOWABLE EXPANSION STRESS RANGE.**

Self-springing occurs only when the system is subjected to higher temperatures. For the expected strain (expected expansion per unit length), if the modulus of elasticity at this high temperature is used to back calculate stress, the stress value will be lower than when it is calculated using modulus of elasticity value at lower temperature (cold condition). That is, calculated stress value is higher when material properties in cold condition are used. This provides a built in safety in design. Hence the stresses are calculated using the **cold modulus of elasticity.** This is a very important point to note. Actual stresses under hot condition would be less than the calculated stresses.

When difference in elastic modulus within a piping system will significantly affect the stress distribution, the resulting displacement stress shall be computed based on the actual elastic modulus and then multiplied by the ratio of the elastic modulus at the ambient temperature to the modulus used in the analysis.

There are other failure modes that could affect the piping system. They include buckling, stress corrosion and brittle fracture. These topics are not ~~correctly~~ currently considered in the piping code. The effects of these must be considered by the Piping Engineer while selecting the materials or restraining the piping system.

2.4.3 ALLOWABLE STRESS RANGE

The failure modes that the piping code addresses are excessive plastic deformation or bursting; plastic instability or incremental collapse due to cycling in the plastic range and fatigue which may be developed in a system as its temperature is raised from the lowest to the highest that it will experience in service or when it is shut down. Each of this failure, modes is caused by a different type of stress and loading. However 'Fatigue failure' is recognized by the code as the most likely mode of failure of the component and place the limit on the maximum stress which may be developed in a system as temperature is raised from

lowest to highest that will experience in service or when it is shut down.

This piping "Shakedown" is also known as self-springing and can be represented as shown in the following sketch (Fig. 2.10).

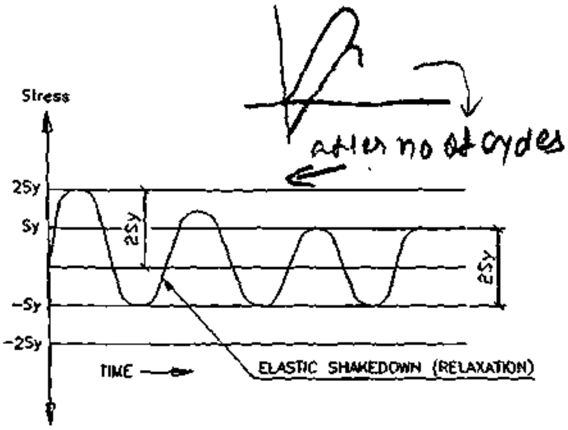


Fig. 2.10

The maximum stress range may be set to 2 times the yield stress, more accurately the run of hot and cold yield stresses, in order to ensure eventual elastic cycling within the bounds of allowable stress. Incorporating a factor of safety, this can be represented by the following equation

$S_E <= F (S_{Yc} + S_{Yh})$

where, S_E – Expansion stress range
F – Factor of Safety
S_{Yc} – Yield stress at installed temperature
S_{Yh} – Yield stress at operating temperature

For materials below the creep ranges, the allowable stresses are 62.5% of the yield stress, so that a conservative estimate of the limit of the bending stress at which plastic flow starts at an elevated temperature is 1.6 (100/62.5) times the allowable stress and by the same reasoning, 1.6 Sc will be the stress at which flow would take place at the minimum temperature. Hence the sum of these stresses represents the MAXIMUM STRESS RANGE to which the system would be subjected to, without the flow occurring in either hot or cold condition. Therefore,

$S_{max} = 1.6\ Sc + 1.6\ Sh = 1.6\ (Sc + Sh)$

But the American design codes ASME B 31.1 and B 31.3 limit the stress range to 78% of the yield stress, which gives a total stress range of

cold = install temp.

$S_{Allowable} = 1.6 \times 0.78\ (Sc + Sh)$
$= 1.25\ (Sc + Sh)$

hot = Design/operating temp

From this total stress range 1 Sh is reserved for the longitudinal stresses developed due to loading such as pressure, weight and other sustained loading, giving the allowable stress range for flexibility as (bcoz, we consider it during sustained load calculation)

$S_{Allowable} = 1.25\ Sc + 0.25\ Sh$

The above value does not consider the excessive cyclic conditions. The code allows it by multiplying by a stress range reduction factor. Accordingly, ASME B 31.1 in clause 102.3.2(c) and ASME 31.3 in clause 302.3.5 specify the Allowable Expansion Stress Range as :

$S_A = f(1.25 S_c + 0.25 S_h)$

where,

S_A = Allowable Expansion Stress Range

S_c = Basic allowable stress at minimum metal temperature during the displacement cycle under analysis

S_h = Basic allowable stress at maximum metal temperature during the displacement cycle under analysis

[The value of S_c and S_h are available in Table A1 of the Code]

f = Stress range reduction factor for displacement cycle conditions for the total number of cycles over the expected life

The factor 'f' has a value of 1.0 for situation where total number of cycles is 7000 or less. This represents one cycle per day for nearly 20 years, which is a common design parameter. Further, if we look at endurance curve for carbon steel and low alloy steel available in the ASME Section VIII Division 2, Pressure Vessel Code, it can be seen that at some point in the vicinity of 7000 cycles, the Sc + Sh limitation intersects the fatigue curve.

The code gives the value of 'f' in the table 302.3.5 (B 31.3) and 102.3.2 (c) (B 31.1) as follows:

Stress Range Reduction Factor f

Cycles N	Factor f
7,000 or less	1.0
over 7,000 to 14,000	0.9
over 14,000 to 22,000	0.8
over 22,000 to 45,000	0.7
over 45,000 to 1,00,000	0.6
over 1,00,000 to 2,00,000	0.5
over 2,00,000 to 7,00,000	0.4
over 7,00,000 to 20,00,000	0.3

This applies essentially to non-corroded piping. Corrosion can decrease the cycle life. Therefore, corrosion resistant material should be considered where large number of stress cycle is anticipated.

f can also be calculated by the equation

$$f = 6.0\,(N)^{-0.2} \le 1.0$$

where,

N = equivalent number of full displacement cycles during the expected service life of piping system [code cautions the designer that the fatigue life of materials operated at elevated temperature may be reduced]

ASME B 31.3- 2004 Edition has revised the Stress Range Reduction Factor ' f ' as per the graph given in Fig 302.3.5., wherein fm , the maximum value of stress range factor is given as 1.2 for ferrous materials with specified minimum tensile strength ≤ 75 Ksi (517 M pa) and at metal temperatures ≤ 700°F (371°C) .

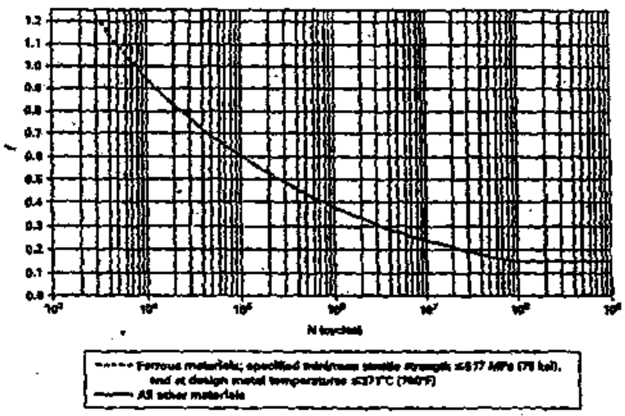


Fig. 302.3.5 Stress Range Reduction factor, f

When the computed stress range varies, whether from thermal expansion or other conditions, S_E is defined as the greater computed displacement stress range. The value of N is such cases can be calculated by the equation

For graph of 'F' see ASME Section-8, Div-2

$N = N_E + \Sigma[\gamma_i^5 N_i]$ for i=1,2.......n

where,

N_E = number of cycles of maximum computed displacement stress range, S_E

$\gamma_i = S_i / S_E$

S_i = any computed displacement stress range smaller than S_E

N_i = Number of cycles associated with displacement stress range S_i

2.4.4 EFFECT OF SUSTAINED LOADS ON FATIGUE STRENGTH

If the alternating stress is plotted against the cycle to failure, it can be seen that the mean stress has an effect on the endurance strength of the material. As the mean stress increases, the maximum permissible absolute stress (S_a + S_m) increases, while the permissible alternating stress decreases. The relation between the allowable alternative stress and the average stress follows the Soderberg line, which correlates fairly well with test data of ductile materials.

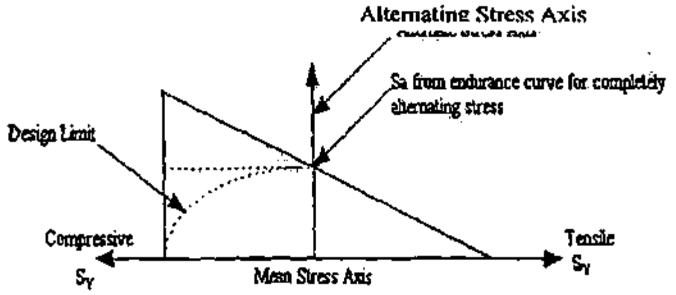


Fig 2.10

The equation for the Soderberg line is

S_a (Allowed) = S_a(for R=1) x (1-S_m/S_{Yield})

where,

$R = S_{min} / S_{max}$

$S_a = S_{max} - S_{min} / 2$

$S_m = S_{max} + S_{min} / 2$

During the development of the ASME BPV code section III for nuclear piping analysis, the special committee to review the Code Stress Basis concluded that the required adjustments to a strain – controlled fatigue data curve based on zero mean stress occur only for the number of cycles in the range 50,000 to 1,00,000 for carbon steels and low alloy steels and are insignificant to austenitic stainless steels and Nickel – Chrome – Iron alloys. Since most of the plant piping come under this material of construction and the cycles of operation comes much fewer than 50,000 cycles in the life of the plant, the effect of mean stress on fatigue life are negligible for piping materials with UTS below 1,00,000 psi. For high strength bolting materials where UTS is greater than 1,00,000 psi the mean stress can have considerable effect on the fatigue strength and should be considered when performing fatigue analysis.

For piping analysis the effect of Soderberg line on fatigue allowable is implemented in a conservative matter. The code 31.1 and 31.3 covers it up in the following manner.

When the basic allowable stress at maximum expected metal temperature ('S_h') is greater than the sum of the longitudinal stresses due to pressure, weight and other sustained loading ('S_L') the difference between them may be added to the term 0.25 S_h in the equation for S_A.

In that case the allowable stress range will be

$S_A = f\,[1.25(S_c + S_h) - S_L]$

Code ASME B31.1 covers these aspects in clause 102.3.2(c). The difference between B31.3 and B31.1 is that the stress reduction factor indicated in Table 302.3.5 covers upto

20,00,000 cycles during the span of the system, whereas Table 102.3.2 (c) in B31.1 indicates this value for 1,00,000 cycles and over. The factors remain the same in both the cases.

ASME B 31.3 - 2004 Edition in Appendix P provides alternative rules for evaluating the stress range in piping systems. It considers stress at operating conditions, including both displacement and sustained loads rather than displacement stress range only. This is more comprehensive than that specified in the text of the code and is more suitable for computer analysis of piping system, including non linear effects such as pipe lifting off of supports.

Let us consider a typical example for calculation of S_A

A pipe supplies Dowtherm to the limpet of a reactor, which is operated on a batch process with a 4-hour cycle every 24 hours. The Dowtherm temperature is 315°C (600°F) and pipe material is ASTM A 106 Gr. B. Design life of plant considered 20 years.

Allowable stress at ambient S_c = 20,000 psi

Allowable stress at Max. metal temp.

S_h = 17,300 psi

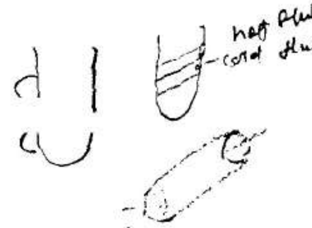
Number of cycles $= \frac{24}{4} \times 365 \times 20$

$= 43,800$ (total)

The stress range reduction factor = 0.7

hence,

$$S_A = f\ (1.25\ S_c + 0.25\ S_h)$$
$$= 0.7\ (1.25 \times 20,000 + 0.25 \times 17,300)$$
$$= 20527 \text{ psi } (1443 \text{ Kg./cm}^2)$$



3.0 LIMITING VALUES OF TERMINAL FORCES AND MOMENTS

It has been indicated earlier that in case of strain sensitive equipment, the main considerations for flexibility are the forces and moments, which a pipe imposes on the equipments to which it is connected. The maximum permitted values of forces and moments will vary with the type of equipment. These are established by the regulatory codes based on which equipment is designed / manufactured. Or else, the manufacturers of the equipment set these values to ensure safe operation of the equipment. The following are the regulatory codes referred by the Piping Engineer.

1. Centrifugal pumps - API 610/ ISO 5199 (Refinery & petrochemicals)
2. Positive displacement pumps - API 676
3. Centrifugal compressors - API 617
4. Reciprocating compressors - API 618
5. Steam turbines - NEMA SM 23 (National electric manufacturing association)
6. Air cooled heat exchangers - API 661/ISO 13706
7. Shell and tube heat exchangers - Manf. Specific.
8. Fired heaters - Manf. Specific.
9. Flat bottom Welded Storage Tanks – API650
10. For other static equipment such as Reactors, vessels and tanks interaction with the fabrication engineer is required to establish that

— half of pipe is used to heat & cool the vessel — Limpet coil

— allowable stress ... beyond allowable stress

the local stress developed due to nozzle loadings are within the acceptable limits. However the values given under section 3.10 can be taken as a guide.

3.1 Centrifugal Pumps

The forces and moments acting on the pump flanges due to pipe loads can cause misalignment of the pump and driver shafts, deformation and overstressing of pump casing or overstressing of fixing bolts between the pump and the base plate.

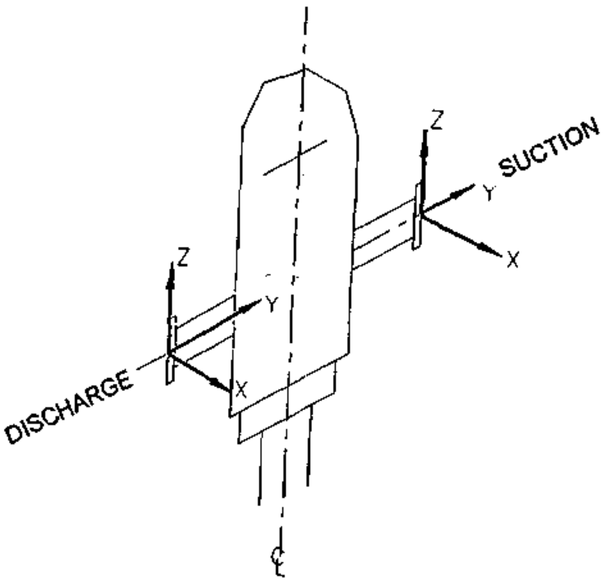
The American Petroleum Institute Standard 610 covering "Centrifugal Pumps for General Refinery Service" is the one, which specifies these limits for pumps. The code specifies the criteria for piping design as Appendix F. The allowable external nozzle forces and moments are tabulated in table 2 for nozzle sizes from 2" NB to 16" NB for pumps with casing constructed out of steel or alloy steel.

Table 3.1: NOZZLE LOADING AS PER API 610

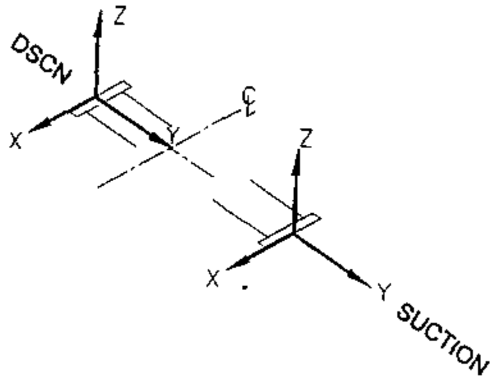
FORCE/MOMENT	NORMAL SIZE OF NOZZLE FLANGE IN INCHES								
	2	**3**	**4**	**6**	**8**	**10**	**12**	**14**	**16**
Each Top Nozzle									
F_X	160	240	320	560	850	1200	1500	1600	1900
F_Y	200	300	400	700	1100	1500	1800	2000	2300
F_Z	130	200	260	460	700	1000	1200	1300	1500
F_R	290	430	570	1010	1560	2200	2600	2900	3300
Each Side Nozzle									
F_X	160	240	320	560	850	1200	1500	1600	1900
F_Y	130	200	260	460	700	1000	1200	1300	1500
F_Z	200	300	400	700	1100	1500	1800	2000	2300
F_R	290	430	570	1010	1560	2200	2600	2900	3300
Each End Nozzle									
F_X	200	300	400	700	1100	1500	1800	2000	2300
F_Y	130	200	260	460	700	1000	1200	1300	1500
F_Z	160	240	320	560	850	1200	1500	1600	1900
F_R	290	430	570	1010	1560	2200	2600	2900	3300
Each Nozzle									
M_X	340	700	980	1700	2600	3700	4500	4700	5400
M_Y	260	530	740	1300	1900	2800	3400	3500	4000
M_Z	170	350	500	870	1300	1800	2200	2300	2700
M_R	460	950	1330	2310	3500	5000	6100	6300	7200

F is Force in pounds ; M is Moment in foot pounds ; R is the resultant ; X, Y, Z : Orientation of Nozzle Loads. API 610 specifies that the pump casing should be designed to withstand double the forces and moments as above. The piping configuration that produces loads and moments outside the above range is also acceptable provided the conditions as specified in Appendix F of the above code are satisfied. For direction of forces and moments see Fig. 3.1

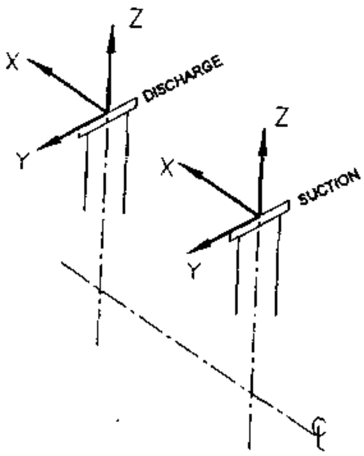
$F_R = \sqrt{F_x^2+F_y^2+F_z^2}$ (Resultant)



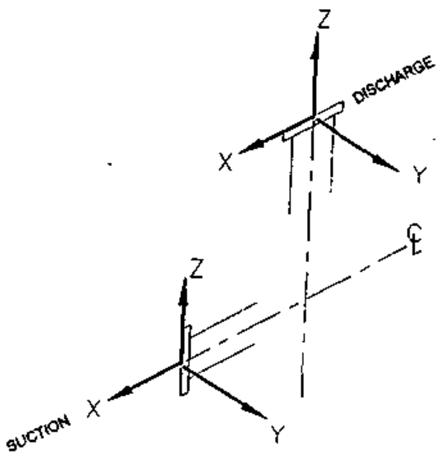
VERTICAL PUMP



HORIZONTAL PUMP WITH SIDE SUCTION SIDE DISCHARGE



HORIZONTAL PUMP WITH TOP SUCTION TOP DISCHARGE

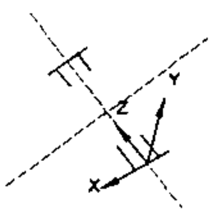


HORIZONTAL PUMP WITH END SUCTION TOP DISCHARGE

Fig.3.1

ISO 5 199 – 2 002(E) in Annex B gives the allowable values of forces and moments on the pump nozzle. The basic values given in table B3 should be multiplied by corresponding coefficient given in table B5 or B6. This is based on the study and tests undertaken within EUROPUMP (European Association of Pump Manufacturers) together with the support of piping specialists.

3.2 Positive Displacement Pumps - Rotary



The American Petroleum Institute Standard 676 specifies in clause 2.4 the limiting values for the Rotary Positive Displacement Pumps with Alloy Steel or Steel Castings at inlet and outlet nozzles as :

F_x	=	75 D lbs (430 D Newtons) inch mm	M_x	=	125 D ft.lbs (2350 D Nmm)
F_y	=	75 D lbs (430 D Newtons)	M_y	=	125 D ft lbs (2350 D Nmm)
F_z	=	75 D lbs (430 D Newtons)	M_z	=	125 D ft lbs (2350 D Nmm)

where D is the nominal diameter of nozzle in inches.

3.3 Centrifugal Compressors

API 617 "Centrifugal Compressors for General Refinery Service" has been specifying that the compressors shall be designed to withstand external forces and moments on each nozzle at least 1.85 times the value calculated in accordance with NEMA - SM 23.

Experience has shown that there has be not been uniform interpretation of "1.85 times NEMA". Therefore API 617 has modified the equations and attached in Appendix G the formulae for the calculation of the acceptable forces and moments.

The forces and moments acting on compressor(s) due to the inlet pipe and discharge pipe connections are:

Limit-1 check

1. The total resultant force and total resultant moment imposed on the Compressor at any connection must not exceed the following:

$3F + M \leq 927D$ or

$$F \leq \frac{927\,D - M}{3}$$

$F + 1.09\,M <= 5.41\,D$ in SI units

F = Resultant force (lbs/Newton)

M = Resultant Moment (ft.lbs/Newton meters)

D = Pipe size of the connection (IPS) in inches/millimeters upto (8) inches (200 mm) in diameter.

For sizes greate than this, use a value of D equal to (16+IPS)/3 inches or (400+D)/3 in mm.

Limit-2 check

2. The combined resultant of the forces and moments of the inlet side and discharge connections **resolved at the centre line of**

the discharge connection must not exceed the following two conditions.

These resultants must not exceed

$$F_c = \frac{462D_c - M_c}{2} \text{ in USCS}$$

$F_c = 40.4D_c - 1.64M_c$ in SI Units

F_c = Combined resultanat of inlet side and discharge forces in pounds/Newtons

M_c = Combined resultant of inlet side and discharge moments resulting from forces in ft lbs/Newton meters

D_c = Diameter (in inches) of a circular opening equal to the total area of inlet side and discharge opening upto a value of nine (9)inches (230 mm) in diameter. For values beyond this use value of

$$Dc = \frac{(18 + \text{Equivalent Diameter})}{3} \text{ inches}$$

$$= \frac{460 + \text{Equivalent Diameter}}{3} \text{ in mm.}$$

limit-3

The components of these resultants shall not exceed

in USCS

$F_y = 231\ D_c$ $M_y = 231\ D_c$

$F_z = 185\ D_c$ $M_z = 231\ D_c$

$F_x = 92\ D_c$ $M_x = 462\ D_c$

in MKS units

$F_y = 40.5\ D_c$ $M_y = 12.3\ D_c$

$F_z = 32.4\ D_c$ $M_z = 12.3\ D_c$

$F_x = 16.1\ D_c$ $M_x = 24.6\ D_c$

where,

F_y = Vertical component of F_c

F_z = Horizontal components of F_c at right angles to compressor shaft

F_x = Horizontal component of F_c Parallel to compressor shaft

M_x = Component of M_c in a hoizontal plane parallel to Compressor shaft

M_y = Component of M_c in a vertical plane

M_z = Component of M_c in a horizontal plane at right angles to the compressor shaft

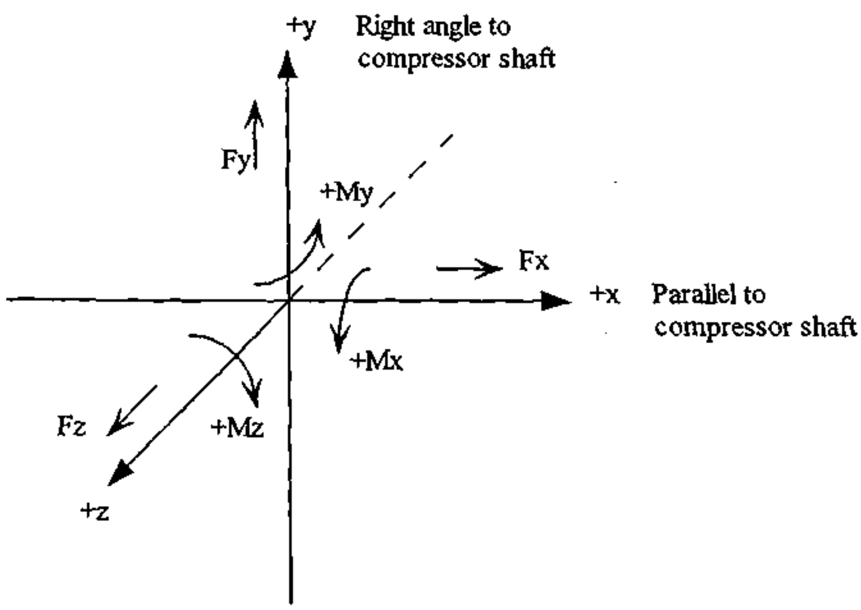
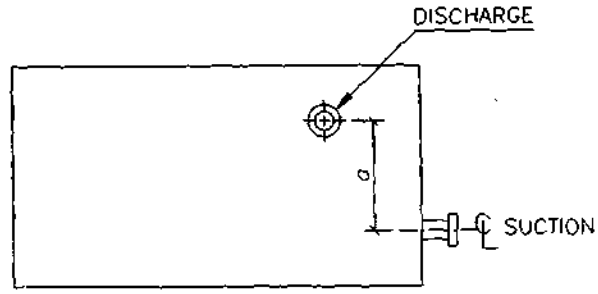
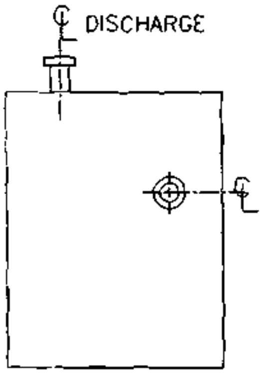


Fig.3.2: Components of forces and moments on compressor connection

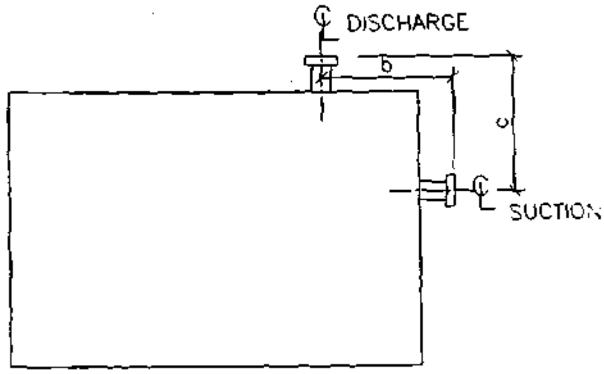
find combined area & then from area find equivalent dia.



PLAN
Fig. 3.3



END VIEW
Fig. 3.4



ELEVATION
Fig. 3.5

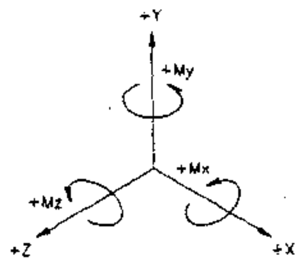


Fig. 3.6

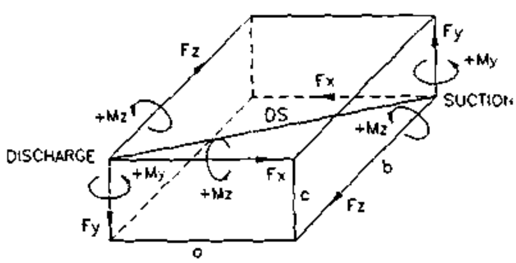


Fig. 3.7

3. These values of allowable forces and moments pertain to the compressor structure only. They do not pertain to the forces and moments in the connecting piping flanges and flange bolting which should not exceed the allowable stress as defined by applicable codes and regulatory bodies.

Forces on smaller connections are to be transferred along with moments to larger connection to analyze the compressor for

limit

Limit-2 check

↳ checking of transfer force & moments from suction to discharge

resultant forces and moments. ***But, the transfer of forces will generate additional transfer moments, which are added to the total of moments to give resultant moments.***

3.4 Reciprocating Compressors

API 618 "Reciprocating Compressor for General Refinery Service" do not specify the limit on the allowable forces and moments in the code. The vendor shall specify the forces and moments for each nozzle in the tabular form. However, the values as per API 610 can be considered for guidance.

3.5 Steam Turbines

NEMA – SM 23 requires that the forces and moments acting on steam turbines due to the steam inlet, extraction, and exhaust connections should be evaluated by simple set of force/moment calculation similar to centrifugal compressors. These computations shall be done as below.

1. The total resultant force and total resultant moment imposed on the turbine at any connection should not exceed the values calculated as per the following equation.

$$3\,F + M < 500\,D \quad \text{or} \quad F < \frac{500D - M}{3}$$

where,

F = Resultant force (lbs) including pressure forces where unrestrained expansion joints are used at the connection except on vertical exhausts. Full vacuum load is allowed on vertical down exhaust flanges.

$$F = \sqrt{F_x^2 + F_y^2 + F_z^2}$$

M = Resultant moment in foot-pounds,

$$M = \sqrt{M_x^2 + M_y^2 + M_z^2}$$

D = Nominal pipe size of the connection in inches up to 8 inches in diameter.

For sizes greater than this, use a value of

$$D \text{ (in inches)} = \frac{(16 + \text{IPS}) \text{ Inches}}{3}$$

2. The combined resultants of the forces and moments of the inlet, extraction, and exhaust connections, resolved at the centerline of the exhaust connection should not exceed the following two conditions.

These resultants shall not exceed:

$$F_c = \frac{250\,D_c - M_c}{2}$$

where,

F_c = Combined resultant of inlet, extraction, and exhaust forces, in lbs.

M_c = Combined resultant of inlet, extraction, and exhaust moments, and moments resulting from forces, in ft lbs.

D_c = Diameter (in inches) of a circular opening equal to the total areas of the inlet, extraction, and exhaust openings up to a value of nine inches in diameter.

For values beyond this, use a value of D_c (in inches) equal to:

$$\frac{(18 + \text{Equivalent diameter in inches})}{3}$$

The components of these resultants should not exceed:

$F_x = 50\,D_c$ $\quad$ $M_x = 250\,D_c$

$F_y = 125D_c$ $\quad$ $M_y = 125\,D_c$

$F_z = 100\ D_c$ $\qquad M_z = 125\ D_c$

The components are as follows:

F_x = Horizontal components of F_c parallel to the turbine shaft.

F_y = Vertical component of F_c

F_z = Horizontal component of F_c at right angles to the turbine shaft.

M_x = Component of M_c around the horizontal axis parallel to the turbine shaft

M_y = Component of M_c around the vertical axis

M_z = Component of M_c around the horizontal axis at right angles to the turbine shaft.

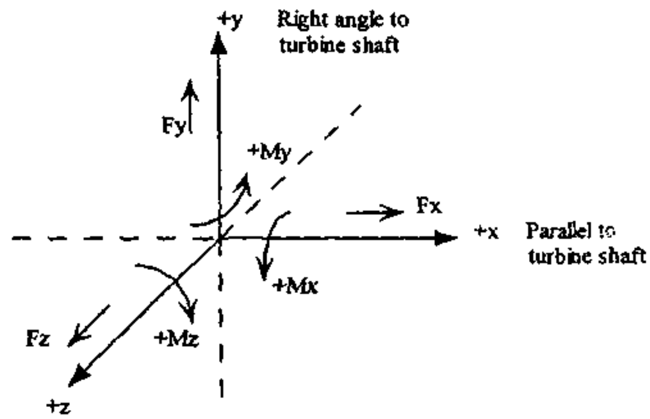


Fig.3.8: Components of forces and moments on turbine connection

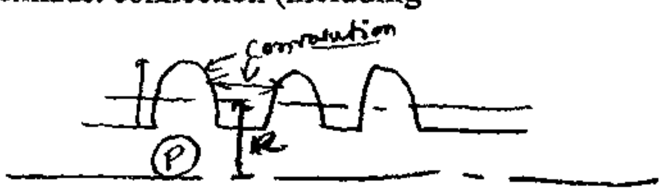
3. For installation of turbines with a vertical exhaust and an unrestrained expansion joint at the exhaust, an additional amount of force caused by pressure loading is allowed. (This additional force is perpendicular to the face of the exhaust flange and is deemed to act at its centre), For this tye of application, calculate the vertical force component on the exhaust connection excluding pressure loading. Compare this with one sixth of the pressure loading on the exhaust. Use the larger of these two numbers for vertical force component on the exhaust connection in making calculations outlined in 1 and 2.

The force caused by the pressure loading on the exhaust is allowed in addition to the values established by the foregoing up to a maximum value of vertical force in pounds on the exhaust connection (including pressure loading) of 15 ½ times the exhaust area in square inches.

4. *These values of allowable force and moment pertain to the turbine structure only. They do not pertain to the forces and moments in the connecting piping, flange, and flange bolting, which should not exceed the allowable stress as defined by applicable codes and explanatory notes.*

3.6 Air Cooled Heat Exchangers

The American Petroleum Institute Standard 661 for "Air Cooled Heat Exchangers for General Refinery Services" covers the allowable loads on the vertical, colinear nozzles found in most single multibundled air-cooled heat Exchangers.



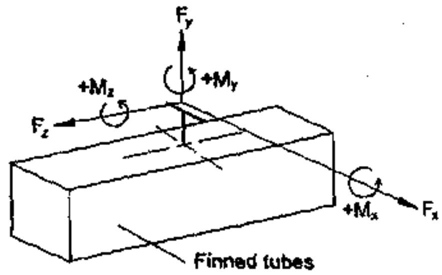


Fig.3.9: Direction of Forces & Moments on the Nozzle

API 661 has the following two requirements.

3.6.1 Each nozzle in corroded condition shall be capable of withstanding the simultaneous application of the following moments and forces.

Table 3.2: Nozzle loading as per API 661 / ISO 13706

Nozzle size	Forces in lbs (N)			Moments in ft lbs (Nm)		
NB mm (In)	Fx	Fy	Fz	Mx	My	Mz
40(1.5)	150(670)	230(1020)	150(670)	80(110)	110(150)	80(110)
50(2)	230(1020)	300(1330)	230(1020)	110(150)	180(240)	110(150)
80(3)	450(2000)	380(1690)	450(2000)	300(410)	450(610)	300(410)
100(4)	750(3340)	600(2670)	750(3340)	600(810)	900(1220)	600(810)
150(6)	900(4000)	1130(5030)	1130(5030)	1580(2140)	2250(3050)	1200(1630)
200(8)	1280(5690)	3000(13340)	1800(8010)	2250(3050)	4500(6100)	1650(2240)
250(10)	1500(6670)	3000(13340)	2250(10010)	3000(4070)	4500(6100)	1880(2550)
300(12)	1880(8360)	3000(13340)	3000(13340)	3750(5080)	4500(6100)	2250(3050)
350(14)	2250(10010)	3750(16680)	3750(16680)	4500(6100)	5250(7120)	2630(3570)

3.6.2 The design of each fixed or floating header, the design of fixed headers to side frames, and the design of other support members shall ensure that the simultaneous application (sum) of all nozzle loadings on a single header will cause no damage. The components of the nozzle loadings on a single header shall not exceed the following values.

M_x = 4500 ft lb (6100 Nm)
M_y = 6000 ft lb (8130 Nm)
M_z = 3000 ft lb (4070 Nm)
F_x = 2250 lb (10010 Nm)
F_y = 4500 lb (20020 Nm)
F_z = 3750 lb (16680 Nm)

Note :- The application of the moments and forces shown in table will cause movement that will tend to reduce the loads to the values given.

3.7 Shell & Tube Type Heat Exchangers

The designer has to set the limiting values or to check the vessel connections for the nozzle loading imposed by the connected piping.

The rough guide generally followed is: -

Resultant Maximum Force	: 200 lb./in NB of nozzle
Bending Moment	: Equivalent to bending stress in standard schedule pipe between 4000 to 5000lbs./in^2

3.8 Fired Heaters

The limiting values for forces and moments should be laid down by the manufacturer. Restrictions are applied on nozzle rotations also in this case to take care of the clearances between the tube and refractory lining. The thumb rule used is :

Forces = 200 to 300 lb/in. nominal bore of nozzle
Moments - Equivalent to Sh/4
Nozzle Rotation - From 1/2° to 1°

3.9 Flat bottom Welded Storage Tanks to API 650

The design of the piping system connected to thin walled, large diameter cylindrical vertical flat bottom storage tanks pose a problem in the analysis of the interface between the piping system and the tank nozzle. The parameters to be considered are the stiffness of the tank shell, the radial deflection and the meridional rotation of the shell opening at the nozzle connection resulting from the static head, pressure and uniform or differential temperature between the shell and the bottom. Although three primary forces and moments may be imparted by the piping on to the shell connection, only the radial thrust (F_R) and two moments i.e. the longitudinal moment (M_L) and the circumferential moment (M_C) are significant causes of shell deformation.

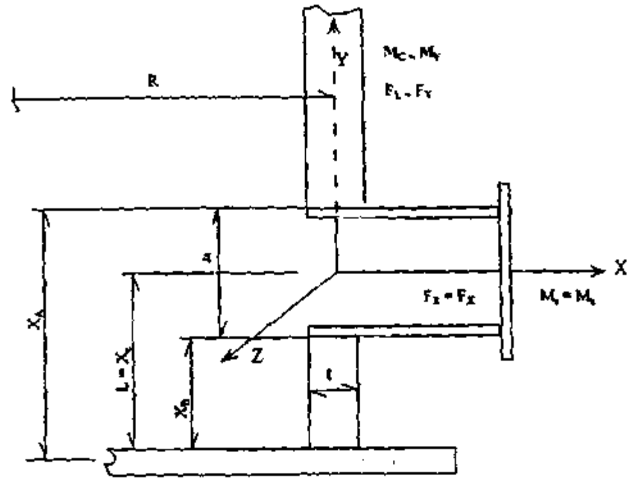


Fig.3.10: Storage Tank Nozzle Details

a = Outside radius of the nozzle in mm(in)
F_R = Radial Thrust applied in N (lbf)
F_P = Pressure head at the nozzle $\pi a^2 P$
L = Vertical distance from centre line of nozzle to tank bottom in mm(in)
t = shell thickness at the nozzle connection in mm(in)
P = Pressure from liquid head in Mpa (lbf/in^2)
$X_A = L + a$ in mm (in)
$X_B = L - a$ in mm (in)
$X_C = L$ in mm (in)
Y_C = Coefficient determined from Fig. P-4B (API650)
Y_F, Y_L = Coefficient determined from Fig. P - 4A (API650)
$\lambda = a/(Rt)^{0.5}$
M_C = Circumferential moment at the nozzle from the piping system in N-mm (in-lbf)
M_L = Longitudinal moment at the nozzle from the piping system in N-mm (in-lbf)

Appendix P of API 650 establishes minimum recommendations for the design of storage tank opening subjected to external piping loads. This is recommended only for tanks larger than 36M (120 ft) in diameter and is considered as an accepted practice for the piping connection at the lower half of the bottom shell course.

welding Research Council
WRC 107

The following are the steps involved in the determination of allowable loads.

a) Determine the non dimensional quantities $X_A/(Rt)^{0.5}$, $X_B/(Rt)^{0.5}$ and $X_C/(Rt)^{0.5}$ for the nozzle connection under consideration.

b) Layout two sets of orthogonal axes on graph paper and label abscissas and ordinates as specified in 'e' below.

c) Layout two sets of orthogonal axes on graph paper and label abscissas and ordinates as specified in 'f' below.

d) Using the values of F_R, M_L and M_C obtained from the piping analysis determine the factors
$(\lambda/2Y_F)(F_R/F_P)$
$(\lambda/aY_L)(M_L/F_P)$
and $(\lambda/aY_C)(M_C/F_P)$

e) Plot point $(\lambda/2Y_F)(F_R/F_P)$, $(\lambda/aY_L)(M_L/F_P)$ on the nomogram with X axis and Y axis respectively.

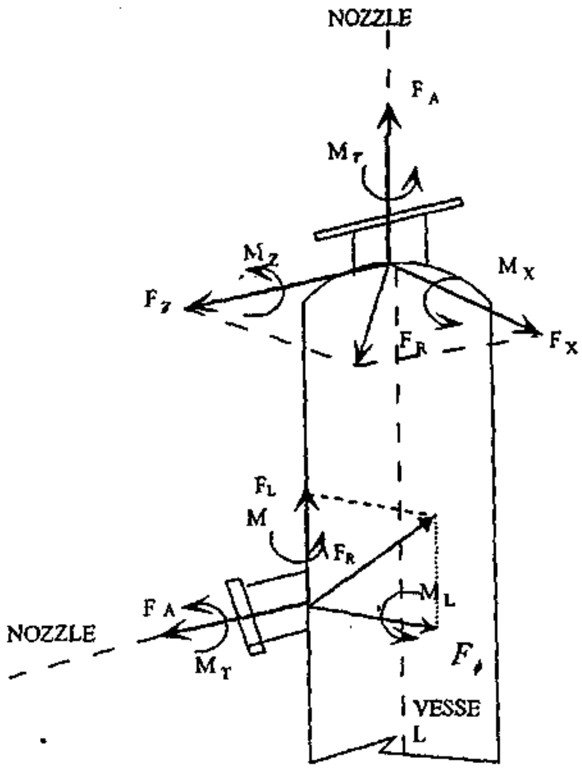
f) Plot point $(\lambda/2Y_F)(F_R/F_P)$, $(\lambda/aY_C)(M_C/F_P)$ on the nomogram with X axis and Y axis respectively.

g) Construct boundaries as lines at 45 degree angles between abscissa and ordinates passing through the calculated value.

h) The external piping loads F_R, M_L and M_C are acceptable if both points determined from b and c above lie within boundaries of the nomogram constructed for that particular nozzle.

3.10 Static Equipment such as columns, reactors, tanks and vessels

Each nozzle, 2" NB and larger, for columns, drums and shell & tube heat exchangers constructed out of steel or alloy is recommended to be designed to withstand forces and moments from the thermal expansion and sustained loading from the piping as per the following criteria. These forces and moments shall be considered to be acting at the intersection of nozzle and shell in he corroded condition. A total of 7000 full temperature cycles shall be considered for the expected life of the equipment.



3.11 Orientation of Forces & Moments on Vertical Equipment

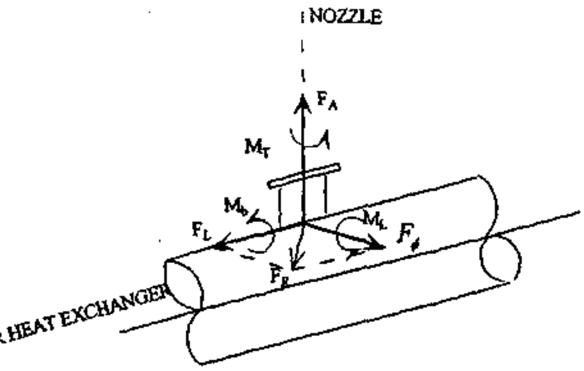


Fig 3.12 Orientation of Forces and Moments on Horizontal Equipment

3.10.1 Nozzle to Shell or channels

a) Moments

i) Longitudinal bending moment
$M_L = \beta\ 130\ D^2$ Nm

ii) Circumferential bending moment
$M_\phi = \beta\ 100\ D^2$ Nm

iii) Resultant bending moment
$M_b = (M_L{}^2 + M_\phi{}^2)^{1/2}$
$= \beta\ 164\ D^2$ Nm

iv) Torsional & moment
$M_t = \beta\ 150\ D^2$ Nm

b) Forces

i) Axial force in plane of Flange
$F_L = \beta 2000D$ N

ii) Tangential force in plane of Flange
$F_\phi = \beta 1500D$ N

iii) Resultant shear force
$F_R = (F_L{}^2 + F_\phi{}^2)^{1/2}$
$= \beta\ 2500D$ N

iv) Radial Tensile or compressive Force
$F_A = \beta\ 2000D$ N

3.10.2 Nozzles to Formed Heads

a) Moment

i) Resultant Bending moment
$M_b = \beta\ 164\ D^2$ Nm
Where M_b is the resultant of the components M_X and M_Z

ii) Torsional Moment
$M_t = \beta\ 150\ D^2$ Nm

b) Forces

i) Resultant shear force
$F_R = \beta 2500D$ N where F_A is the resultant of the components F_X and F_Z

ii) Radial Tensile or compressive force
$F_A = \beta 2000D$ N
Where β is as per the table 3.3 and D is the nominal diameter in inches. The orientation of the forces and moments shall be as per Fig 3.11 & 3.12. These loadings shall be considered as being caused by 67 % thermal and 33 % dead weight load.

Flange Rating		β Valve	
ANSI Class	DIN	Heat Exchangers	Columns and Drums
150 #	PN 10 & 16	0.75	0.6
300 #	PN 25 & 40	0.75	0.7
600 #	PN 64 & 100	1.25	0.8
900 #	PN 160	3.00	1.8
1500 #	PN 250 & 320	4.00	3.0
2500 #	PN 400	5.60	3.3

Table 3.3

3.10.3 The local stress intensity at the nozzle connection due to attachment of piping can be computed using the welding research council bulletin 107 setting the limitations as.

a) The local sustained stress intensity at the nozzle connection should be less than 0.5 sm
b) The sum of local sustained stress intensity and the local expansion stress intensity at the nozzle connection must

be less than 2 sm

where, sm is the allowable stress intensity for the material at the operating temperature.

4.0 Data required for flexibility calculations

The following data will be required for the flexibility calculations if it is carried out manually or by the use of software. It is therefore prudent to have this ready before starting.
The direction of coordinates are fixed as below:

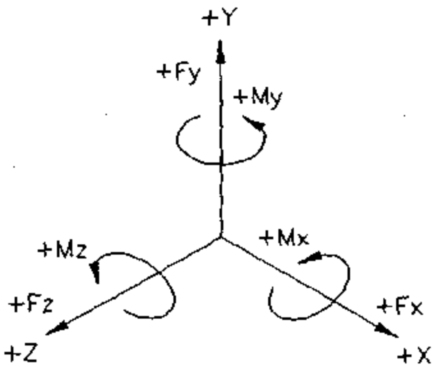


Fig. 4.1

1. Code of Practice
2. Basic Material of Construction of Pipe
3. Ambient / Installation temperature
4. Number of Thermal Cases
5. Flexibility Temperature (See Note)
6. Design Pressure
7. Outside diameter of Pipe
8. Type of construction of pipe
9. Nominal Thickness of Pipe
10. Manufacturing tolerance
11. Corrosion allowance
12. Pipe Weight
13. Insulation Weight
14. Specific Gravity of Contents
15. Young's Modulus at Ambient/Installation Temperature
16. Young's Modulus at Flexibility Temperature
17. Thermal Expansion at Flexibility Temperature
18. Allowable stress at Ambient/ Installation temperature
19. Allowable stress at flexibility temperature
20. Bend radius and type of bend
21. Branch connection type
22. Weight of attachments - Valves and Specialties
23. Terminal movements with directions

<u>Note</u>: The Code states that the design temperature shall be assumed to be same as the fluid temperature unless calculations or test supports the use of other data.

5.0 METHODS OF FLEXIBILITY ANALYSIS

5.1 Check As Per Clause 119.7.1/ 319.4.1 of the Code

Clause 119.7.1(A3)/ 319.4.1(c) specifies that no formal analysis is required in systems which are of uniform size, have no more than two points of fixation, no intermediate restraints and fall within the empirical equation.

$$\frac{DY}{(L-U)^2} \le K_1$$ (hardly used)

where,
D = the outside diameter of pipe

SS expands more than CS

System wise checking is done not line wise

In inch (or mm)

Y = Resultant of total displacement strains In inch (or mm) to be absorbed by the Piping system.

L = Developed length between the anchors In ft. or (m)

U = Anchor distance, straight line between anchors in ft. or (m)

$K1 = 30\ S_A/E_a$ in USCS

$= 208300\ S_A/E_a$ in SI units

Where E_a is the Modulus of Elasticity at the installation temperature

and S_A is the allowable stress range

<u>Example:</u> Let us apply the above check for the following system.

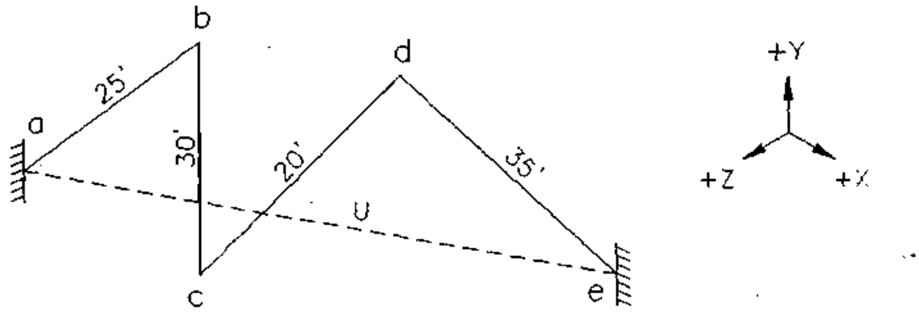


Fig. 5.1

Pipe - 6" (150 mm NB) Sch. 40 carbon steel to ASTM A106 Gr. B
Design Temperature - 400 °F (204°C)

Step 1

To establish the anchor to anchor distance U

Total length in X direction = 35'

Total length in Y direction = 30'

Total length in Z direction = 25' + 20'
= 45'

Step 2

To determine value of L.

$L = |x|+|y|+|z| = 35 + 30 + 45 = 110$ ft.

Step 3

To calculate resultant total displacement Y

From Appendix C, ASME B 31.3
Linear Expansion between 70° F and 400°F.

e = 2.7" /100 ft. (coeff. of thermal expansion)

$$\Delta x = \frac{2.7 \times 35}{100} = 0.945"$$

$$\Delta y = \frac{2.7 \times 30}{100} = 0.810"$$

$$\Delta z = \frac{2.7 \times 45}{100} = 1.215"$$

$$Y = \sqrt{\Delta x^2 + \Delta y^2 + \Delta z^2}$$

$$= \sqrt{0.945^2 + 0.810^2 + 1.215^2}$$

$$= 1.739"$$

Step 4

$$K = \frac{DY}{(L-U)^2}$$

$$= \frac{6.625 \times 1.739}{(110-64.42)^2}$$

$$= 0.0055$$

To calculate K_1

Allowable stress at 400°F = 20,000 Psi
Allowable stress at 70°F = 20,000 Psi
SA = f (1.25 Sc + 0.25 Sh)
= 1 (1.25×2000 + 0.25×20,000)
= 30,000 psi
Ea = 29.3×106 Psi

$$K1 = 30S_A / E_a \frac{30 \times 3 \times 10^4}{29.3 \times 106} = 0.0307$$

since $K \leq K_1$ the configuration is safe.

Please note that no general proof can be offered that this equation will yield accurate and conservative results. It is not applicable to systems used under severe cyclic conditions. There is no assurance that the terminal reactions will be acceptably low, even if the system satisfies the above equation.

5.2 Guided Cantilever Method

Suppose that we have two vessels T1 and T2 say 20 m apart and we have to run the pipe from T1 and T2 between two nozzles at the same elevation. Obviously the most economical way of doing this, purely from the performance aspect, is to join them by a straight pipe as shown in Fig. 5.2.

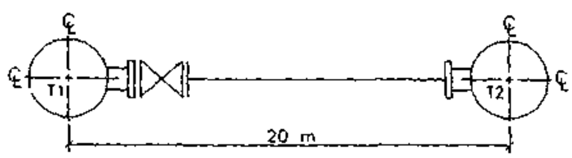


Fig. 5.2

Now further, suppose that everything is in carbon steel and the vessel T1 has its temperature raised to 200°C. When the valve is opened, there will be expansion in the connecting pipeline, which can be calculated as below.

Expansion of carbon steel from 21°C to 200°C = 2.2 mm/ m

(Refer Appendix C ASME B 31.3)

Total Expansion = 20 x 2.2 = 44 mm
To absorb this expansion, one of the following two things can happen.

1. As the pipe expands it will dent the sides of the vessel as in Fig.5.3

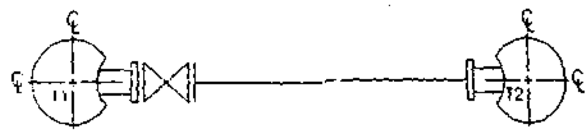


Fig. 5.3

2. The pipe will buckle as shown in Fig. 5.4 if the vessels are of large diameter and, therefore, thick and the pipe is small.

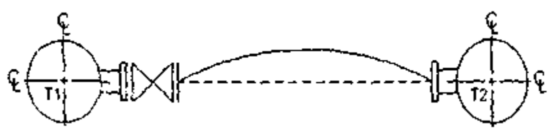


Fig. 5.4

However, if the equipment is laid out differently, it will be possible to run the pipe in two different sections at right angles to each other as shown in Fig. 5.5.

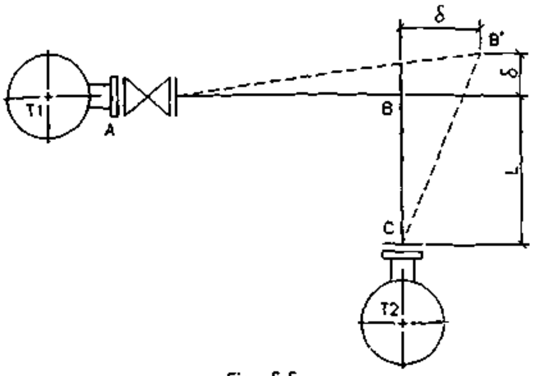


Fig. 5.5

With this configuration of piping, as the point B moves out to B', it is able to bend the leg BC to position B'C.

It is simple to calculate length L of BC which will allow expansion " δ " to be absorbed while the stresses are restricted to a given value and this is the simplest concept of all in the field of flexibility analysis, namely "Minimum Length".

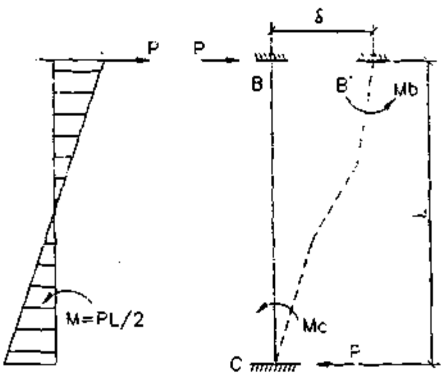


Fig. 5.6

when the pipe bends as shown by the dotted line, i.e. B'C, it is referred as "Guided cantilever "

As per Elastic Theory,

$$\delta = \frac{Pl^3}{12EI}$$

where,

δ=Movementin inches

P=Force required to bend BC in lbs

l=Length of BC in inches

E=Young's Modulus in lbs/in^2

I=Moment of inertia about bending axis in in^4

If L is length of BC in ft. ($l = 12$ L)

$$\delta = \frac{144\,PL^3}{EI}$$

Hence,

$$P = \frac{EI\delta}{144\,L^3} \text{ above}$$

Maximum bending moment at B or C $= \pm PL/2$ $= M$ ft.lbs.

Maximum bending stress $f = \dfrac{MY \times 12}{I}$ lbs/ in^2

$$Y = \frac{\text{OD of pipe}}{2}$$

$$f = \frac{12MY}{I}$$

$$= \frac{12\,PL}{I\;\,2} \times \frac{D}{2}$$

Substituting $P = \dfrac{EI\delta}{144\,L^3}$

$$f = \frac{12}{I} \times \frac{EI\delta}{144\,L^3} \times \frac{L}{2} \times \frac{D}{2}$$

i.e. $f = \dfrac{DE\delta}{48L^2}$

$$L = \sqrt{\frac{DE\delta}{48f}}$$

By putting the value of stress range calculated, as discussed earlier and the modulus of elasticity of the material, we can arrive at the "Minimum Leg Length".

e.g. :- In the previous layout if we restrict the stress at 16,000 psi and consider modulus of elasticity of carbon steel as 29.5 x 10^6 psi and assume the pipe size as 6" NB (6.625" OD)

Expansion of piping between T1 and T2,

$\delta = 0.87"$ (22 mm)

$$L = \sqrt{\frac{DE\delta}{48f}}$$

$$= \sqrt{\frac{6.625 \times 29500000 \times 0.87}{16000 \times 48}}$$

=14.88 ft.(4.54 m)

This indicates that the length BC should not be less than 4.6 m.

We can also calculate the stress developed in such a system of known dimensions of leg BC by the same method.

$$\delta = \frac{Pl^3}{12EI}$$

$$\text{Hence } P = \frac{12\,EI\delta}{l^3}$$

$$M = \frac{Pl}{2}$$

$$= \frac{12\,E\,I\,\delta \times l}{l^3 \quad 2} = \frac{6\,EI\delta}{l^2}$$

$$F = \frac{M}{Z} = \frac{6\,EI\,\delta}{l^2\,Z}$$

$$R = \frac{I}{Z} \;;\; Z = \frac{I}{R}$$

$$\text{Solving for } S_E = \frac{6\,E\,R\,\delta}{l^2}$$

Where R = Outer radius of pipe, inches
I = Moment of inertia of cross section, in^4
Z = Section modulus, in^3
E = Modulus of elasticity, lbs/in^2
l = Length, inches

In Fig.5.5 if the vessels are arranged in such a way that AB and BC are equal and 10 M each, then the stress developed can be calculated as

l = AB = BC = 10 m = 394 inches
E = 29.5 x 10^6 lbs/ in^2
R = 6.625/2 inches
δ = 1.73/2 inches

$$S_E = \frac{6 \times 29.5 \times 10^6 \times 6.625 \times 1.73}{(394)^2 \times 2 \times 2}$$

= 3267 psi

5.3 Piping Elements - Their Individual Effects

Let us analyze each of the piping elements to see how it contributes to flexibility.

5.3.1 STRAIGHT PIPE : FLEXIBILITY IN TORSION

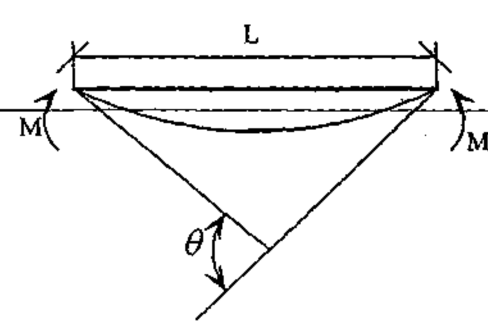


Fig 5.7

If a bending moment M is applied to the end of a straight piece of pipe, it behaves as a uniform beam and exhibits change of slope from end to end, as given by the expression.

$$\theta = \frac{ML}{EI}$$

Where,

θ = Angle, radians

M = Bending moment, in lbs (mm – N)

E = Young's Modulus, lbs/in^2 (KPa)

I = Moment of Inertia, in^4 (mm^4)

L = Length, inches (mm)

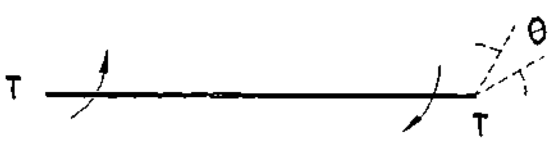


Fig 5.8

If the same pipe is subjected to a constant twisting moment, the rotation of one end relative to the other end is given by :

$$\theta = \frac{TL}{GJ}$$

Where,

θ = Angle of twist in radians

T = Torsion moment in – lbs (mm – N)

L = Length, inches (mm)

G = Modulus of rigidity, lbs/in^2 (KPa) $= \frac{E}{2(1+m)}$

J = Polar moment of inertia, in^4 (mm^4)

poisson's Ratio = 0.3

It can be shown that for metals

G = E / 2.6 and

J = 2 x I for circular cross section

Hence,

$$\theta = \frac{T \times L}{E / 2.6 \times 2I}$$

$$\theta = 1.3 \frac{TL}{EI}$$

This result is important as far as piping routings in three planes are concerned. It shows that a given length of pipe can give 30% more rotation if the moment from the adjacent legs produces torsion instead of bending. This can create moments, which should also be kept in limits.

There are piping components other than straight pipe, which are required to make a complete piping system. These are elbows, tees, reducers, valves, etc. The knowledge of individual effects on the flexibility and the stresses in each element is essential to analyze a piping system close to its true nature of behavior.

5.3.2 ELBOWS:

Early calculation on the flexibility analysis containing elbows proved that the structural engineering theory and the results of experiments did not agree well. Practical piping system was more flexible than predicted and the discrepancy was due to the flexibility of elbows.

The first theoretical analysis of the behavior of pipe bends when subjected to a bending moment was made by Theodore Von Karman, who showed that when a curved pipe is subjected to a bending moment in its own plane, the circular section gets flattened and this results in increased flexibility. This was further

developed by Hovgarrd, Berkins, Vigners & Markel.

Flexibility Factor

The ratio of the flexibility of a bend to that of a straight pipe having the same length and cross section is known as its "Flexibility Factor", usually denoted by the alphabet "k".

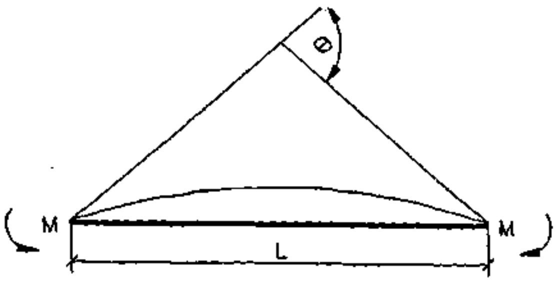


Fig. 5.9

$$\theta = \frac{ML}{EI}$$

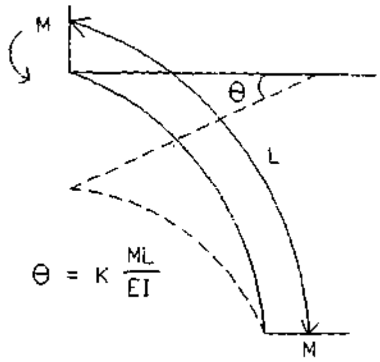


Fig. 5.10

Let us consider how flattening of the cross section occurs

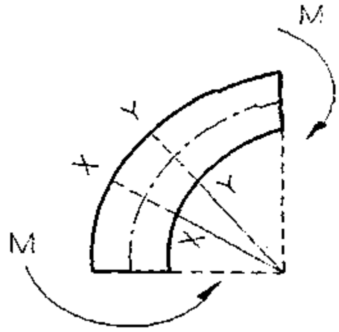


Fig. 5.11

When bending moment M is applied as shown, tensile stresses are developed on the outer fibres and compressive stresses on the inner ones.

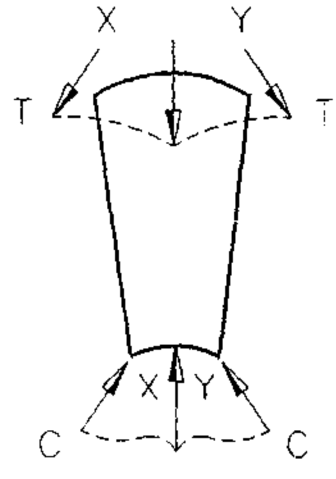


Fig. 5.12

Let us consider a thin slice taken between two radial planes "XX" and "YY". (see Fig. 5.12).

The resultant effect of the tensile load "T" in the outer fibres is an inward radial load on the element. Similarly the resultant of the compressive loads "C" in the inner fibres is an inward radial load on the element.

If we view the slice as a cross section, and draw a loading diagram for the ring, we arrive at the situation shown below. Under the applied loading the ring

flattens into an ellipse with its major axis horizontal (see Fig. 5.13.a)

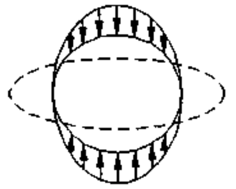


Fig. 5.13a

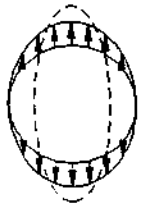


Fig. 5.13b

If the bending moment is reversed, the tensile and compressive forces will also get reversed and cross section gets elongated instead of getting flattened. (See Fig. 5.13.b)

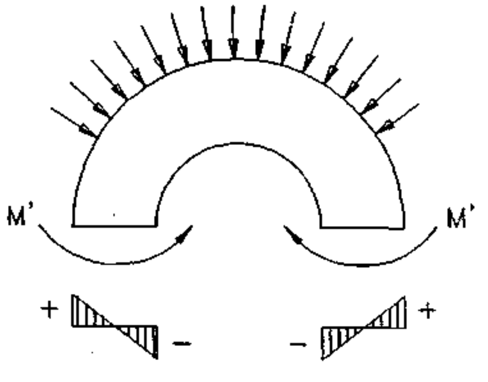


Fig. 5.14 Circumferential Stress in Pipe wall

If the element is analyzed in more detail, it is seen that the flattening produces bending moments in the rings, which are maximum at the ends. These moments produce a stress which varies from tension to compression through the thickness of pipe wall and which is circumferential in direction. If we consider one half of the ring, the stress system gets illustrated as above.

These circumferential stresses due to bending moment M can be many times the value of MY/I obtained by bending theory of structural members. The factor by which the circumferential stresses exceed the longitudinal stresses in the bend is called the "Stress Intensification Factor" called S.I.F. It can be defined as the ratio of the actual bending stress for a moment applied to nominal section.

The effect of the existence of these circumferential stresses is that when elbow is subjected to repeated "in-plane" bending, it ultimately develops a fatigue crack along its sides.

When we take additional benefit by Flexibility factor due to flattening of elbows, consideration should be given to the induced circumferential stresses by multiplying the stresses at the bends due to overall bending moment by the appropriate "Stress Intensification Factor".

Appendix D of ASME B 31.1 & 31.3 tabulates the expressions to be used for calculating the Flexibility Factor and Stress Intensification Factor. The parameter used for the calculation of these factors is called the "Flexibility Characteristic" denoted by letter "h"

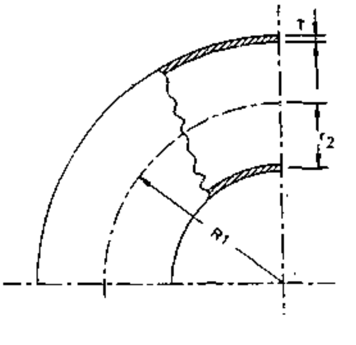


Fig. 5.15

Flexibility Characteristic $h = \dfrac{TR1}{(r_2)^2}$

T =Wall thickness, inches (mm)

R_1 =Mean radius of bend, inches (mm)

r_2 =Mean radius of pipe, inches (mm)

Using this parameter, code indicates that

The flexibility factor = k = $1.65/h$
Inplane S.I.F. = i_i = $0.9/h^{2/3}$
Outplane S.I.F. = I_o = $0.75/h^{2/3}$

When any problem is analyzed, the following considerations are made.

a) The Flexibility Factor applies to bending in any plane.

b) The stress intensification factor is greater for "inplane" bending than for "outplane" bending. ASME B 31.3 permits the use of inplane SIF for any plane whereas B 31. 1 does not separate out these two.

Virtual Length

The product of length of arc centre line and Flexibility factor is referred as "Virtual length" of a bend and these are considerable while analyzing thin walled large diameter pipes.

Mitre Bends

In case of Mitre Bends an equivalent bend radius is used in the equation to calculate 'h'. The equivalent bend radius (Re) is estimated by

$$Re = r_2(1 + 0.5s / r_2 \cot\theta)$$

for closely spaced

$$Re = r_2(1 + \cot\theta)$$

mitres

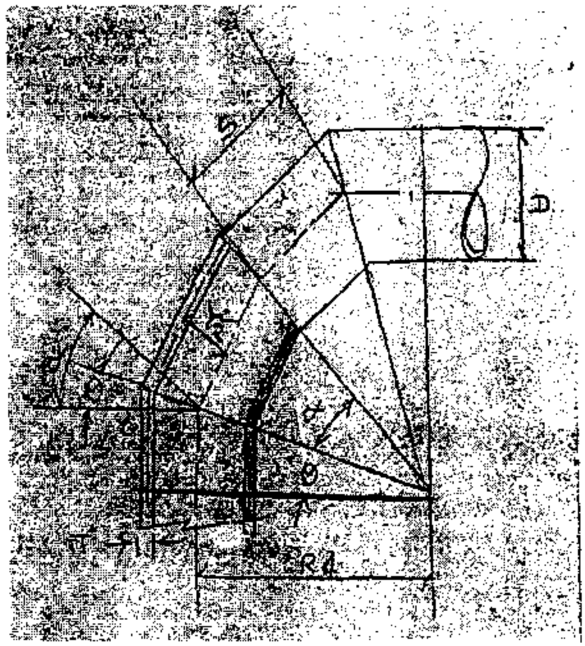
for widely spaced mitres
(4 piece & above)

where,

S = mitre spacing at centre line, inches (mm)

θ = One half of angle between cuts

r_2 = mean radius of pipe, inches (mm)



5.3.3 TEES

As far as branch off is concerned the flexibility factor is 1 and the stress intensification factor can be calculated based on the specific formula adapted by code ASME B31.1 / B31.3 and given in Appendix D. These vary depending on the type of branch connection.

The unreinforced fabricated tee is modelled using same formula for widely spaced miter bend with single miter i.e. half angle of 45°. This produces the flexibility characteristic of

$$h = T / r_2$$

For buttweld tees, Markyl adapted bend equation with equivalent radius (Re) and equivalent thickness (Te).

$$h = c(Te\,Re / r_2{}^2)$$

where,

c = ratio of tee to pipe section modulii.
= $(Te/T)^{3/2}$ as recommended by ARC Markyl.

Te = Equivalent pipe wall thickness inches (mm)

= 1.60T as recommended by ARC Markyl

Re = Equivalent bending radius inches (mm)

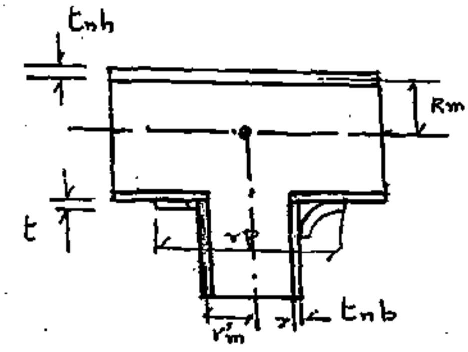
= 1.35 r_2 as recommended by ARC Markyl

Substituting these values in the expression for h

$$h = (Te/T)^{3/2}(Te1.35r_2/r_2^{\,2})$$

$$h = 4.4T/r_2$$

As far as the stub connections are concerned, the major problem is in the out of plane bending moment on the header. Stresses due to these moments can never be predicted from the size on size tests. Errors due to these moments can be non-conservative as much as a factor of two to three. Further when r_2 / R ratio is very small the branch connection has little impact on the header and the calculated stress could be unreasonably large by using large SIF. It has been pointed out by R. W. Schneider of Bonney Forge that the highest stress intensification factor occurs when the ratio of branch to header is 0.7 at which the non-conservation is of the order of two.



Main / Branch >>>>>>>>>>>>>>>>>>			4"/1"	4"/2"	6"/1"	6"/2"	6"/3"	8"/1"	8"/2"	8"/3"	8"/4"
Mean Radius of Run pipe	Rm	mm	54.14	54.14	80.585	80.585	80.585	105.46	105.46	105.46	105.46
Nominal Thickness of Run pipe	tnh	mm	6.02	6.02	7.11	7.11	7.11	8.18	8.18	8.18	8.18
Mean Radius of Branch pipe	r'm	mm	14.425	27.30	14.425	27.30	41.705	14.425	27.30	41.705	54.14
Nominal Thickness of Branch pipe	tnb	mm	4.55	5.54	4.55	5.54	5.49	4.55	5.54	5.49	6.02
Outside Radius of Branch Reinforcement	rp	mm	16.7	30.15	16.7	30.15	44.45	16.7	30.15	44.45	57.15
Stress Intensification Factor 1.5((Rm/tnh)^(2/3))*(r'm/Rm)^(1/2)*(tnb/tnh)*(r'm/rp)	SIF		2.1000	3.0503	1.7701	3.1210	3.9447	1.4085	2.5946	3.2000	4.1100
Ratio of Moment of Inertia of (MAIN / BRANCH)			137.75	12.01	644.00	88.95	12.34	1061.70	174.36	36.00	13.50

Main / Branch >>>>>>>>>>>>>>>>>>			10"/1"	10"/2"	10"/3"	10"/4"	12"/1"	12"/2"	12"/3"	12"/4"	12"/6"
Mean Radius of Run pipe	Rm	mm	131.015	131.015	131.015	131.02	157.165	157.165	157.165	157.165	157.165
Nominal Thickness of Run pipe	tnh	mm	9.27	9.27	9.27	9.27	9.53	9.53	9.53	9.53	9.53
Mean Radius of Branch pipe	r'm	mm	14.425	27.30	41.705	54.14	14.425	27.30	41.705	54.14	80.585
Nominal Thickness of Branch pipe	tnb	mm	4.55	5.54	5.49	6.02	4.55	5.54	5.49	6.02	7.11
Outside Radius of Branch Reinforcement	rp	mm	16.7	30.15	44.45	57.15	16.7	30.15	44.45	57.15	84.15
Stress Intensification Factor 1.5((Rm/tnh)^(2/3))*(r'm/Rm)^(1/2)*(tnb/tnh)*(r'm/rp)	SIF		1.2049	2.1776	2.7820	3.3714	1.2042	2.1415	2.7690	3.3106	4.3730
Ratio of Moment of Inertia of (MAIN / BRANCH)			4469.80	420.74	88.06	32.50	6044.19	537.40	174.21	64.40	13.72

5.3.4 FLANGES

For flanges also the flexibility factor is 1 and the various types of flanges are considered to have the following Stress Intensification Factors.

S I F for Flanges:

Weld neck flange	1.0
Slip-on flange	1.2
Socket weld flange	1.3
Lap joint flange	1.6
Threaded flange	2.3

The flange when attached to the bend exerts a severe restraint to the flattening of the cross section due to its heavy construction. Hence attachment of the flange to an elbow or a mitre bend reduces the flexibility as well as the stress intensification factor. Flange at both ends of the elbow reduces these factors further.

ASME B 31.3 indicates these correction (reduction) factor as:

$C_1 = h^{1/6}$ for one end flanged

$C_1 = h^{1/3}$ for both ends flanged

Flanges are designed to remain leakfree under hydrostatic test pressure when cold and under operating pressure when hot. The design of flanges does not take into account bending moment in the pipe. However, the flanges transmit some bending moment before they 'open up',

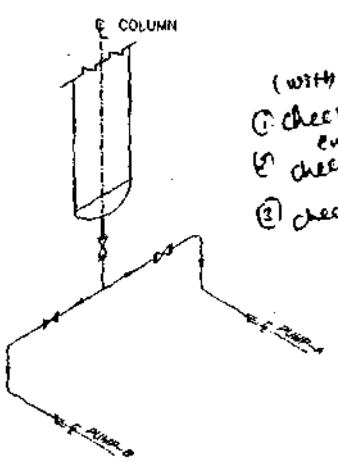
the value being very small. This generates the wire drawing effect on the mating surface of the flange. Hence, additional flexibility is to be considered where a flange joint is located near a point of high bending moment.

5.3.5 REDUCERS

The reducer could be either eccentric or concentric and in both the cases the stress intensification factor and flexibility factor are 1. The overall length is very small compared to the piping and hence the effect of the same is neglected in ASME B31.3. ASME B31.1 indicates the SIF as 2 max for a concentric reducer as per ASME B16.9.

5.3.6 VALVES

Valve is normally considered as a short length of very thick pipe. The effect of temperature when the valve is closed is more significant in the analysis.



(with pump-2 shutdown)
① check 4 pump-1
(with pump-1 shutdown)
② check 4 pump-2
③ check 4 system

Fig. 5.16

5.3.7 EFFECT OF PRESSURE ON SIF AND FLEXIBILITY FACTOR

In large diameter thin walled elbows and bends, pressure can significantly affect the Flexibility Factor 'k' and Stress Intensification Factor 'i'. Hence the correction factor as below should be applied on the values available from the table.

$$\text{Divide 'k' by } \left[1+6\left(\frac{P}{E}\right)\left(\frac{r_2}{T}\right)^{7/3}\left(\frac{R_1}{r_2}\right)^{1/3}\right]$$

$$\text{Divide 'i' by } \left[1+3.25\left(\frac{P}{E}\right)\left(\frac{r_2}{T}\right)^{5/2}\left(\frac{R_1}{r_2}\right)^{2/3}\right]$$

where

T = Nominal wall thickness of fitting, inches (mm)
r_2 = Mean radius of the matching end, inches (mm)
P = Guage pressure psi (KPa)
E = Modulus of Elasticity psi (KPa)
R_1 = Bend radius, inches (mm)

This is called Bourden effect and this stiffening effect of pressure on bends are not considered in ASME B 31.1.

6.0 CODE STRESS EQUATIONS

The stress equations specified in the code are derived from the theoretical and substantiated by investigative work. To make the calculation simpler, the code calculates the stress intensity only for expansion stress, since this load case contains no hoop or radial components.

6.1 ASME B 31.1

ASME B 31.1 specifies under clause 104.8 that to validate a design under the rules of this clause, the complete piping system must be analyzed between anchors for the effects of thermal expansion, weight, other sustained loads and other occasional loads.

6.1.1 STRESS DUE TO SUSTAINED LOADS

The effects of pressure, weight and other sustained mechanical load must meet the requirements of the following equation.

$$S_L = \frac{P\,Do}{4\,t_n} + \frac{0.75\,i\,M_A}{Z} \le S_h$$

in USCS units

$$S_L = \frac{P\,Do}{4\,t_n} + \frac{1000(0.75\,i)\,M_A}{Z} \le S_h$$

in SI units

where

S_L = Sum of the longitudinal stresses due to pressure, weight and other sustained loads, psi (KPa)

i = Stress intensification factor (ref. Appendix D-1)
The product 0.75i shall never be taken as less than 1.

M_A = resultant moment due to weight and sustained loads, in-lb (mm – N)

$$= \sqrt{M_x^2 + M_y^2 + M_z^2}$$

Z = Section Modulus, in^3 (mm^3)

t_n = Nominal Thickness, in (mm)

S_h = Basic allowable stress at the operating temp., psi (KPa)

6.1.2 THERMAL EXPANSION STRESS RANGE

The effects of thermal expansion must meet the requirements of the following equation.

$$S_E = \frac{i\,M_c}{Z} \le S_A + f\,(S_h - S_L)$$

In USCS units

$$S_E = \frac{1000\,i\,M_c}{Z} \le S_A + f\,(S_h - S_L)$$

In SI unit

Where,

S_E = Expansion stress range psi (KPa)

M_c = Range of resultant moments due to thermal expansion, in - lb (mm – N)

$$= \sqrt{M_x^2 + M_y^2 + M_z^2}$$

S_A = Allowable stress range (Ref 2.4.3 above) psi (KPa)

6.1.3 STRESS DUE TO OCCASIONAL LOADS.

The effects of pressure, weights, other sustained loads and occasional loads including earthquake must meet the requirements of the following equation.

$$\frac{P\,Do}{4\,t_n} + \frac{0.75\,i\,M_A}{Z} + \frac{0.75\,i\,M_B}{Z} \le KS_h$$

in USCS units.

$$\frac{P\,Do}{4\,t_n} + \frac{1000(0.75\,i)\,M_A}{Z} + \frac{1000(0.75\,i)\,M_B}{Z} \le KS_h$$

In SI units.

Where

K = 1.15 for occasional loads acting less than 10% of any 24 hr. operating period.

K = 1.2 for occasional loads acting less than 1% of any 24 hr. operating period.

M_B = Resultant moment loading on cross section due to occasional loads. If calculation of moments due to earthquake is required, use only one half of the

earthquake moment range. Effect of anchor displacement due to earthquake may be excluded from the equation if they are covered in Thermal Expansion stress range calculation.

6.2 ASME B 31.3

6.2.1 ASME B 31.3 does not provide an explicit equation for sustained stress calculation, but requires that Piping Engineer should compute the longitudinal stresses due to weight; pressure and other sustained loading and ensure that these do not exceed Sh. The thickness of pipe used in calculating S_L shall be the nominal thickness less the erosion and corrosion allowance. This is calculated by looking at Clause 302.3.5 (c)

$$S_L = \frac{F_{ax}}{A_m} + \frac{\sqrt{(i_i M_i)^2 + (i_o M_o)^2}}{Z} + \frac{PDo}{4t}$$

$\leq S_h$ in USCS units

$$S_L = \frac{F_{ax}}{Am} + \frac{1000\ [(i_i M_i)^2 + (i_o M_o)^2]^{1/2}}{Z}$$

$\leq S_h$ in SI units

where,

S_L = Sum of longitudinal stress due to pressure weight and other sustained loading, psi (KPa)

F_{ax} = Axial force due to sustained (primary) loading, lbs (kg)

A_m = Metal cross sectional area, in^2 (mm^2)

M_i = In-plane bending moment due to sustained (primary) Loading, in-lb (mm–N)

M_o = Out-plane bending moment due to sustained (primary) Loading, in-lb (mm–N)

i_i, i_o = In-plane and out-plane stress intensification factors

S_h = Basic allowable stress at the operating temp., psi (KPa)

6.2.2 THERMAL EXPANSION STRESS RANGE

The computed displacement stress range shall be done as below (Ref. Clause 319.4.4).

(a) The range of bending and torsional stresses shall be computed using the as installed Modulus of Elasticity 'E_a' and then combined as below to determine the computed stress range.

$$S_E = \sqrt{S_b^2 + 4S_t^2}$$

where

S_b = Resultant bending stress, psi (KPa)

S_t = Torsional Stress = Mt/ 2z, psi

1000Mt/ 2z, KPa

M_t = Torsional moment, in - lb (mm - N)

Z = Section Modulus of Pipe, in^3 (mm^3)

Appendix P of ASME B 31.3 – 2004 Edition has modified this clause as;

The stress due to bending, torsion and axial loads shall be computed using the cold modulus of elasticity E_a and then combined in accordance with equation.

$$S_o = \sqrt{(|S_a| + S_b)^2 + 4S_t^2}$$

to determine the operating stress S_o and equation.

$$S_E = \sqrt{(|S_a| + S_b)^2 + 4S_t^2}$$

to determine the maximum operating stress range S_E.

S_{om} is greater of the operating stress S_O, and maximum operating stress range SE. which shall not exceed the allowable stress

$$S_{OA} = 1.25f\,(S_c+S_h)$$

Where S_a = Stress due to axial force
= $i_a F_a / A_p$.

F_a = axial force, including that due to internal pressure.

I_a = axial force stress intensification factor.In absence of more applicable.

A_p = cross sectional area of the pipe.

(b) The resultant bending stress to be used in the above equation for elbows and full size branch connection shall be calculated as follows

$$S_b = \frac{\sqrt{(i_i M_i)^2 + (i_0 M_0)^2}}{Z}$$

$$S_b = \frac{1000\sqrt{(i_i M_i)^2 + (i_0 M_0)^2}}{Z}$$

where

$i_i = in - planeSIF$

$i_o = out - planeSIF$

$M_i = in - planeBendingMoment$

$M_o = out - planeBendingMoment$

$Z = SectionModulusofPipe$

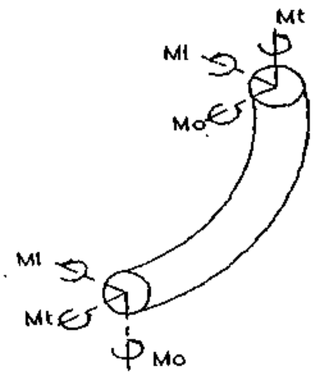


Fig. 6.1 – Moments in Bends

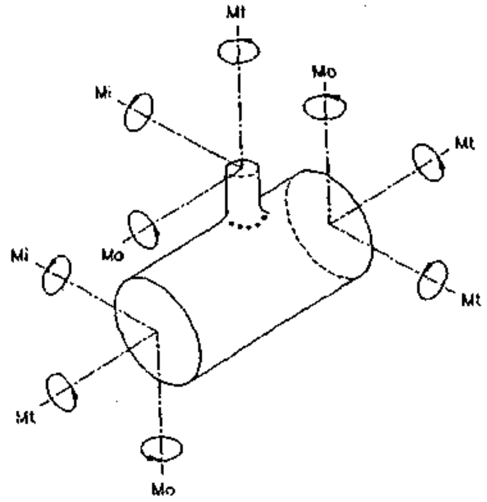


Fig. 6.2 – Moments in Tees

For Reducing outlet branch connections, the equation shall be as follows.

For Header

$$S_b = \frac{\sqrt{(i_i M_i)^2 + (i_0 M_0)^2}}{Z}$$

in USCS units.

$$S_b = \frac{1000\ [\,(i_i M_i)^2 + (i_o M_o)^2\,]^{1/2}}{Z}$$

in SI units.

For Branch

$$S_b = \frac{\sqrt{(i_i M_i)^2 + (i_o M_o)^2}}{Z_e}$$

in USCS units

$$S_b = \frac{1000\left[(i_i M_i)^2 + (i_o M_o)^2\right]^{1/2}}{Z_e}$$

in SI units

where

S_b=Resultant bendingstress

Z_e=effective section modulus of branch

$= \pi r_2^2 T_s$

r_2=mean branch cross - sectional radius

T_s=effective branch wall thicnkess,

lesser of $\bar{T}h$ and $(i_i)(\bar{T}b)$

$\bar{T}h$=Thickness of pipe matching run of tee or header exclusive of renforcement

$\bar{T}b$=Thickness of pipe matching branch

For branch connection, the resultant bending stress needs special care as section modulus Z of header and branch is different.

6.2.3 STRESS DUE TO OCCASIONAL LOADS

ASME B 31.3 do not specifically define the equation for calculating the stresses due to occasional loads. The code, under clause 302.3.6 only states that the sum of longitudinal stresses due to sustained and occasional loads shall not exceed 1.33 times the basic allowable stress. The method adopted is to calculate the sustained and occasional stresses independently and to then add them absolutely.

6.3 COMPARISON OF CODES

Based on the above, we can identify the difference in approach between these two codes

6.3.1 Stress increase due to occasional loads are different in each code.

6.3.2 Allowable stresses are different for each code.

6.3.3 ASME B 31.3 include Fax in the stress calculation due to sustained load where as ASME B 31.1 neglects all forces

6.3.4 ASME B 31.3 calculation methods are undefined for sustained and occasional loads whereas ASME B 31.1 is explicit for the same.

6.3.5 For calculation of stresses due to sustained loads ASME B31.3 neglects torsion where as ASME B31.1 includes it.

6.3.6 ASME B31.1 intensifies torsion where as ASME B 31.3 does not.

7.0 MEANS OF INCREASING FLEXIBILITY

The pipe thickness has no significant effect on bending stress due to thermal expansion but it affects the end reactions in direct ratio. So overstress cannot be nullified by increasing the thickness; on the contrary, this makes the matter worse by increasing the end reactions. This is demonstrated in the following example.

Let us consider two simple cantilever arrangements having the same deflection, pipe size and length but with varying thickness.

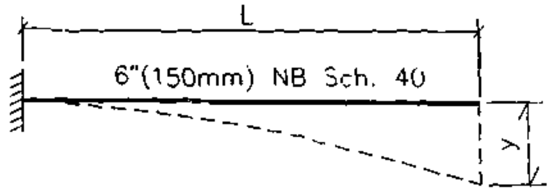


Fig. 7.1

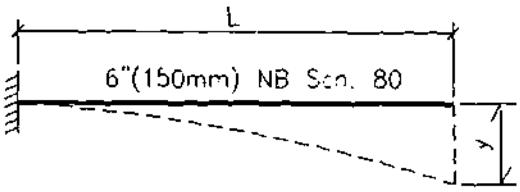


Fig. 7.2

For simple cantilever, the deflection is given by the formula

$$y = \frac{P L^3}{3 E I}$$

$$\text{Hence } P = \frac{3 E y I}{L^3}$$

E, y, L remaining the same, $P = k I$

$$\text{where } k = \frac{3E y}{L^3}$$

For 6"(150 mm) NB Sch. 40 pipe

$I = 1170 \text{ cm}^4$
$Z = 139 \text{ cm}^3$

For 6" (150 mm) NB Sch. 80 pipe

$I = 1686 \text{ cm}^4$
$Z = 200 \text{ cm}^3$

Therefore,

	Sch. 40	Sch. 80
Load P	1170 k	1686 k
Moment M	1170 k L	1686 k L
Stress = M/Z	8.4 k L	8.4 k L

Form the above it can be seen that although the stress remains the same, the forces and moments increase with the increase in thickness of the pipe.

Where the piping system encounters large end reactions or detrimental over strain and it lacks built in changes in the direction to absorb the same, the Piping Engineer should consider adding flexibility by one or more of the following means; addition of bends, loops or offsets, swivel joints, corrugated pipes, expansion joints or other means permitting angular, rotational or axial movements. Suitable anchors shall be provided to resist the end forces.

8.0 COLD SPRING

Piping Engineer may also provide cold cut or cold spring to limit the value of stress and moment.

Cold spring is defined by the code ASME B 31.3 under clause 319.2.4 as the intentional deformation of piping during assembly to produce a desired initial displacement and stress.

No credit for cold spring is permitted in the stress range calculation since the service life of a system is affected more by the range of stress variation than by magnitude of stress at a given time.

ASME B 31.3 gives the formula for calculation of maximum reaction or moment when cold spring is applied to a two anchor piping system in clause 319.5.1 as below.

$$R_m = R\left(1 - \tfrac{2}{3}c\right)\frac{E_m}{E_a}$$

where

To join the above 2 pipes,
the adjustment done/force required, procedure is

R = Estimated instantaneous maximum reaction force or moment at maximum or minimum metal temperature.

R = Range of reaction force or moments derived from flexibility analysis corresponding to the full displacement stress range and based on E.

E_a = Modulus of elasticity at installation temperature.

E_m = Modulus of elasticity at design temperature.

C = Coldspring factor from 0 for no coldspring to 1.0 for 100% coldspring.

The factor 2/3 is based on experience, which shows that specified cold spring

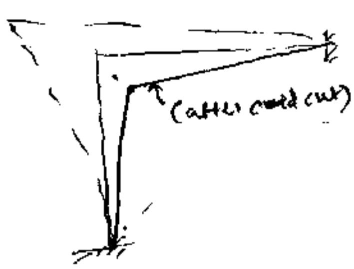
cannot be fully assured even with elaborate precautions.

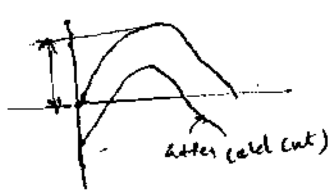
The value of reaction forces or moments at the temperature at which the piping is assembled is given by:

R_a = CR or C_1R which ever is greater

$$C_1 = 1 - \frac{S_h E_a}{S_E E_m}$$

R_a = Estimated instantaneous reaction or moments at the installation temperature.





S_E = computed displacement stress range

S_h = Maximum allowable stress at design temperature

ASME B 31.1 deals with these factors under the clause 119.9 and 119.10.

9.0 SELECTED CHART SOLUTIONS

9.1 The book on Flexibility analysis "Piping Stress Calculations Simplified", by S. W. Spielvogel deals with a number of shapes frequently encountered in practice. This can be used for a quick check on the terminal forces and moments. This method neglects the effect of flexibility of the bends and to that extent the values are over estimated.

The reaction thus computed shall not exceed the limits, which the

attached equipment can safely sustain.

One of the shapes considered is the "Three Dimensional 90° Turns" comprising of three legs, each of which is at right angles to the other two.

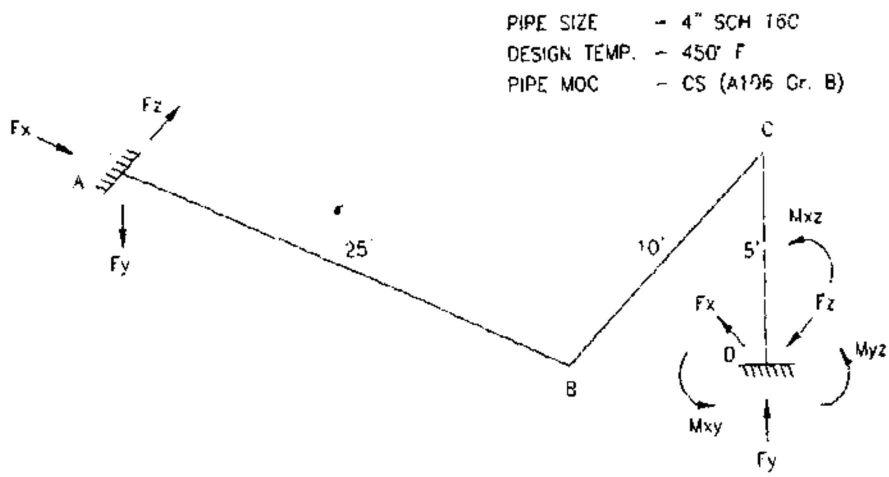


Fig. 9.1

AB = L_1 = 25'
BC = L_2 = 10'
CD = L_3 = 5'
L_1/L_3 = m = 25/5 = 5
L_2/L_3 = n = 10/5 = 2
By referring to chart, we get :

K_b = 8.61 K_t = 4.35

K_x = 1.60 K_y = 0.09 K_z = 0.57

K_{xy} = 1.40 K_{xz} = 1.50 K_{yz} = 0.40

The book gives the following formula

The stresses are evaluated from the equation,

$S = KC\,D/L_3$ lbs/ sq. inch

The forces are evaluated from the equation,

$F = KC\,I/L_3{}^2$ lbs

Moments are evaluated from the equation,

$M = KC\,I/L_3$ ft/lbs

where C is the expansion factor calculated from the expression

$$C = \frac{\text{Expansion in inches}/100 \times Ec}{1728 \times 100}$$

Expansion of C.S. @ 450 °F
= 3.16 inch / 100 ft.

Ec = Cold Modulus of Elasticity
= 27.9×10^6 lbs/sq.in.

Hence, $C = \dfrac{3.16 \times 27.9 \times 10^6}{1728 \times 100} = 510$

I_p for 4" NB Sch 160 pipe = 13.3 in^4.

Allowable stress at installation temp
Sc = 20,000 psi
Allowable stress at design temp
Sh = 19,450 psi

$$\text{Bending Stress} = K_b C \frac{D}{L_3}$$

$$= \frac{8.61 \times 510 \times 4.5}{5}$$

$$= 3952 \text{ lbs./sq.inch}$$

$$\text{Torsional Stress} = K_t C \frac{D}{L_t}$$

$$= \frac{4.35 \times 510 \times 4.5}{5}$$

$$= 1997 \text{lbs./sq.inch}$$

Expansion Stress Range

$= S_E = \sqrt{Sb^2 + 4St^2}$

$= \sqrt{(3952)^2 + (1997)^2}$

$= 5619$ lbs./sq.inch

Allowable stress range = S_A

$= f\,(1.25\,S_C + 0.25\,S_h)$

$= 1\,(1.25 \times 20{,}000 + 0.25 \times 19{,}450)$

$= 29{,}862$ psi

$S_E < S_A$.

Reaction $F_x = K_x C \frac{I_p}{L^3}$

$= \frac{1.6 \times 510 \times 13.3}{5 \times 5}$

$= 434$ lbs.

Reaction $F_y = K_y C \frac{I_p}{L^3}$

$= \frac{0.09 \times 510 \times 13.3}{5 \times 5}$

$= 24$ lbs.

Reaction $F_z = K_z C \frac{I_p}{L^3}$

$= \frac{0.57 \times 510 \times 13.3}{5 \times 5}$

$= 155$ lbs.

Moment $M_{xy} = K_{xy} C \frac{I_p}{L_3}$

$= \frac{1.4 \times 510 \times 13.3}{5}$

$= 1899$ ft.lbs.

Moment $M_{xz} = K_{xz} C \frac{I_p}{L_3}$

$= \frac{1.5 \times 510 \times 13.3}{5}$

$= 2035$ ft.lbs.

Moment $M_{yz} = K_{yz} C \frac{I_p}{L_3}$

$= \frac{0.4 \times 510 \times 13.3}{5}$

$= 543$ ft.lbs.

9.2 TUBE TURN METHOD

The above method (Guided Cantilever Analysis) does not give allowance for the elbow flexibility.

Chart solutions incorporating the flexibility of elbows for certain single plain configurations were produced by M/s. Tube Turns Inc. Charts were developed which unfortunately limit the application to Z, L, U and symmetric expansion loops. Nothing is available for 3 dimensional configurations.

In this method, the expansion stress is calculated by the expression.

$$f = \frac{f_e f_s}{f_1}$$

where,

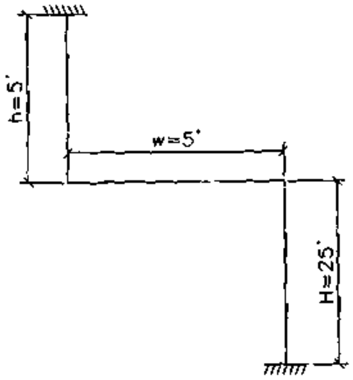
f_e = Expansion Factor allowing for the temperature and material of pipe given in table.

f_s = Shape factor allowing for the ratios of lengths.

f_1 = Factor for effective diameter length, allowing for the excess virtual length (EVL) of the elbow.

These EVL are tabulated in the table.

For example



Pipe size	=	4" std. wt. ASTM A106Gr.B
Operating Temp.	=	450 °F

From table

Effective Elbow Diameter	Dr	=	8.78 inch
Effective Elbow Length	Lr	=	4.7 feet
Length of Short Vertical Leg	: h	=	5'
Length of Long Vertical Leg	: H	=	25'
Length of horizontal offset	: W	=	5'

h/H = 5/25 = 0.20
H/W = 25/5 = 5.00

From Chart, shape factor	f_s	=	1.66
Square corner length		=	h + H + W = l = 35.0'
Sum of elbow lengths		=	2 Lr = 2 x 4.7 = 9.4'
Total effective length		=	l + 2 Lr = L = 44.4'
Effective Diameter length		=	L/Dr = f_l = 44.4/8.78 = 5.06

From Table 1

Expansion factor f_e for 450 °F = 73,000 psi

This can also be calculated by the formula Expansion inches/inch x Young's Modulus i.e

$f_e = E_c \times \Delta_L$

For C.S. $E_c = 27.9 \times 10^6$

Expansion in inches per 100 ft. for C.S. = 3.16 inch from Appendix C. ASME B31.3

Hence,

$$f_e = \frac{3.16 \times 27.9 \times 10^6}{100 \times 12} = 73,470 \text{ psi}$$

Computed Stress Range

$$S_E = \frac{f_e f_s}{f_l}$$

$$= \frac{73000 \times 1.66}{5.06} = 23948 \text{ psi}$$

$$S_A = f(1.25Sc + 0.25Sh)$$

For CS to A106 Gr.B,

S_c = 20,000 psi

S_h = 19,450 psi at 450°F

$$S_A = 1(1.25 \times 20,000 + 0.25 \times 19450)$$

$$= 29,862 \text{ psi}$$

$$S_E < S_A$$

10.0 COMPUTER ANALYSIS

So far the consideration was given only for the calculation of stresses in a pipeline with uniform diameter and supported at two fixed points. This does not represent a life size problem. In a real case, the pipe routings will be connecting to various equipments, guides or restraints will be provided in between anchors and the pipes will be of different diameters and thicknesses. Further one section of the piping may be cold while the other part is subjected to high temperatures.

There is computer software available to handle such complex problems. Some of the software available are: -

1. ADL PIPE
2. AUTOPIPE
3. CAESAR II
4. CAEPIPE
5. PIPEPLUS
6. TRIFLEX
7. Q-FLEX

The pipeline geometry is fed into the system along with all the parameters such as design temperatures, pipe sizes; bend radii, type of branch connections, locations of anchor points and restraints. This is termed as 'Modeling' the problem. The model can be generated by anybody who knows how to prepare the input. The programme executes the solutions. First computer analysis was done in the year 1957.

The analysis of the solutions is the real engineering and is the job of a Piping Engineer.

11.0 ANALYSIS OF REBOILER CONNECTION

Let us analyze the requirements of the flexibility of connecting piping for the vertical thermosyphon reboiler, which is most commonly used. The analysis is restricted to the fixed tube sheet type of heat exchanger for simplicity. Being fixed tube sheet type of heat exchanger, there will be a bellow type expansion joint provided on the shell. This is to accommodate the differential expansion between the reboiler shell and the tubes. This has nothing to do with the flexibility of the piping connection.

Let us first look into the functioning of the system before analyzing the various arrangements of the reboiler supports. The heating fluid in the reboiler shell provides latent heat to the liquid inside the tubes, thereby vaporizing a part of it. The mixture of vapor and liquid returns to the column via vapor return connection. This mixture will be at the same temperature as the liquid drawn from the bottom of the column. When the shell of the column and the tubes of the reboiler are of the same material of construction, then the vertical expansion between the nozzles on the column will be practically same as the expansion between two corresponding nozzles on the reboiler

There can be different ways of attachment of the reboiler to the column. First we will consider the simplest system to analyze as illustrated in the Arrangement-1.

11.1 Arrangement-1 (See Fig. 11.1)

Here the reboiler is supported on its shell from the bottom part of the column as near to the elevation of the vapor return nozzle as possible. In this case the support lugs on the reboiler shell is arranged as near to the top tube sheet as possible. This minimizes the differential expansion between the centre line of the vapor return nozzle on the column and the corresponding reboiler nozzle i.e. between ab and a'b'.

As indicated earlier, when the tubes are of the same material as that of the shell of the column, the differential expansion in vertical plane between the nozzle on the column and those on the reboiler will be negligible in the normal operating conditions.

When the tubes and the column shell have different material of construction, the piping leg 'cd' in Arrangement-1 has to accommodate the differential expansion. For accommodating expansion in the horizontal plane, the reboiler is supported on low friction slide plates, which allow the same to move.

If an operating condition occurs in such a way that the temperature of the reboiler and/or the column varies, then the differential expansion will have to be accommodated by the flexibility in the connecting piping. This condition could occur if the heating of reboiler is started before establishing the level in the column or with the result of a leaky valve in the heating circuit.

As the top nozzle is rigidly attached and normally will be of higher sizes designed for two phase flow, the flexibility can be achieved only by the liquid inlet piping. Hence the analysis has to be done for the upset condition as above and not the normal operation.

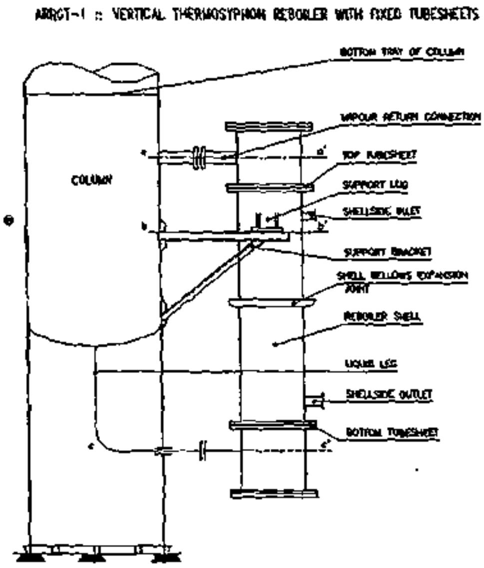


Fig. 11.1

11.2 Arrangement-2 (See Fig. 11.2)

If it so happens that the column shell will not permit the support of reboiler on brackets attached to the shell just below the vapor return nozzle, then the brackets are to be attached to the vessel skirt. This will result in the support bracket at a larger distance from the level of the vapor return nozzle.

In all reboiler analysis problems, the centre line of the vapor return nozzle is considered as the datum and the movements are assessed first on the column side and then on the reboiler side. Applying this to the present problem, the expansion of the reboiler shell between the vapor return nozzle and the support bracket will be more as the shell is at a higher temperature and the tubes and column shell are at the same temperature. This differential will not be possible to be accommodated in the connection between the reboiler outlet and the column, as they are close coupled.

The only solution to this problem is to support the reboiler on the spring supports with the addition of low friction slide plates to allow the reboiler support to

slide. This is as illustrated in the Arrangement - 2.

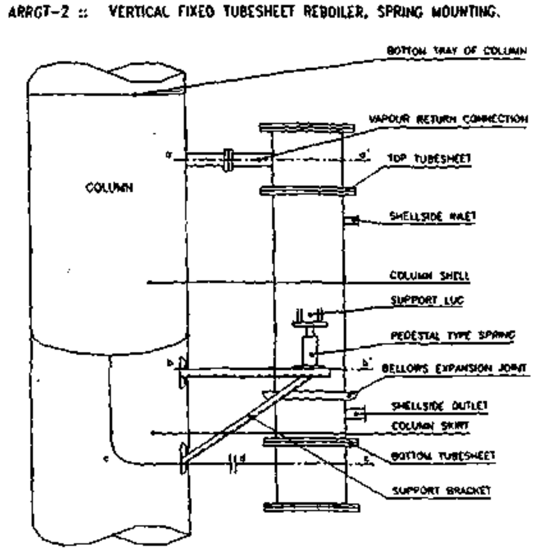


Fig. 11.2

11.3 Arrangement-3 (See Fig. 11.3)

When the reboiler is too heavy to support from the column or the skirt as illustrated above, an independent support has to be provided for support of the reboiler. In this case there will be a considerable offset between the column and the reboiler centrelines. The movements of vapour return nozzle will be at fixed elevation.

The spring support is the only solution and will be as illustrated in Arrangement-3 (See Fig. 11.3)

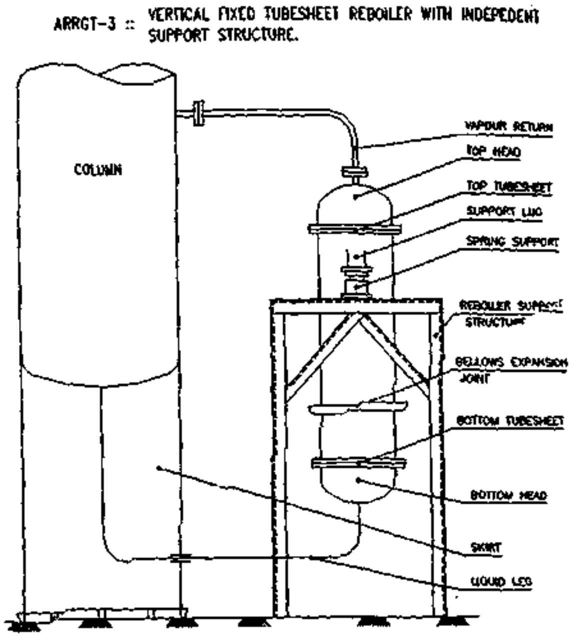


Fig. 11.3

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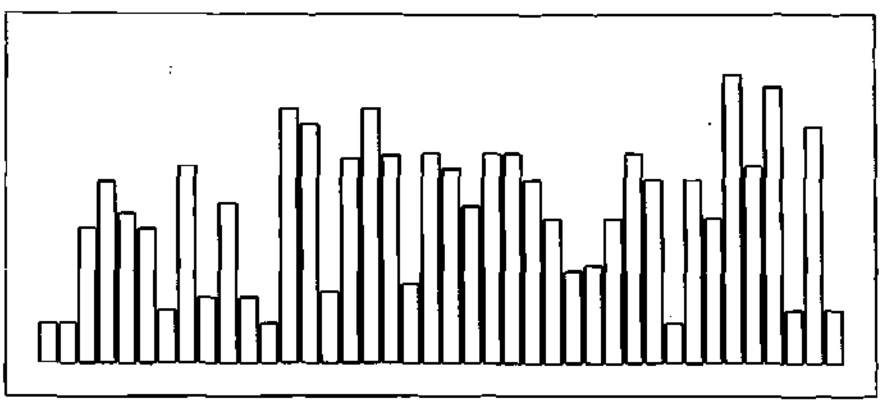
Certificate Course
on
PIPING ENGINEERING

June 12 - 25, 2006

SELECTION OF PIPE SUPPORTS

T. N. Gopinath
Consultant



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Piping Engineering Cell
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Indian Institute of Technology, Bombay
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WEIGHT ANALYSIS AND SELECTION OF PIPE SUPPORTS FOR CRITICAL PIPING

T. N. GOPINATH

1.0 GENERAL

The type and location of supports in a piping system cannot be divorced from the flexibility calculation. The selection and design of pipe supports is an important part of the Piping Engineering of any modern Chemical/ Petrochemical process plant. The problems related with the high pressure/ high temperature piping are critical to the point that the concentrated load imposed by the piping on the building structure and/ or on the equipment nozzles are to be taken into consideration at the early stages of the design. The various stages involved in the design and engineering of pipe supports are presented here in their proper sequence to serve as a good reference document for the Piping Engineer.

The first step is to collect the necessary amount of basic information.
These include

i) A complete set of piping general arrangement drawings.
ii) A complete set of steel and structural drawings including the equipment foundation.
iii) A complete set of drawings showing the location of ventilating ducts, electrical trays, instrument trays etc.
iv) A complete set of piping specification and line list, which includes pipe sizes, material of construction, thickness of insulation, operating temperatures etc.
v) A copy of insulation specification with densities.
vi) A copy of valve and specialty list indicating weights.
vii) The movement of all critical equipment connections such as turbines, compressors, boilers, etc.

On collection of the above data, the steps in which the engineer will apply this basic information are as follows:

i) The determination of support location.
ii) The determination of thermal movement of the piping at each support location.
iii) The calculation of load at each support location.
iv) The selection of the type of support i.e. Anchor, Guide, Rest, Constant or Variable spring etc.
v) Checking the physical interference of the support with structures, trays, ducts equipments etc.

The final step does not need discussion to a great extent. This will be governed solely by the requirement of the individual layout. Steps 1 to 4 will be general and hold good for any installation the following explanation will serve as a guide to any pipe support problem.

The term "Supports" or "Supporting Elements" encompasses the entire range of various methods of carrying the weight of the pipeline and the contents. It therefore includes "hangers" which generally carry the weight from above, with the supporting members being mainly in tension. Likewise, it includes "supports" which on occasion are delineated as those which carry weight from below, with supporting member being in compression.

The code ASME B 31.3 specifies under clause 321.1.1, the objective of the support design as:

Insulation materials – mineral wool
– Calcium silicate (In case of high temp.)

The layout and the design of the piping and its supporting elements shall be directed towards preventing the following.

1. Piping stresses in excess of those permitted in the code.
2. Leakage at joints
3. Excessive thrust and moments on connected equipment (such as pumps and turbines)
4. Excessive stresses in the supporting (or restraining) elements.
5. Resonance with imposed fluid induced vibrations.
6. Excessive interference with thermal expansion and contraction in a piping system, which is otherwise adequately flexible.
7. Unintentional disengagement of piping from its supports
8. Excessive piping sag in systems requiring drainage slope.
9. Excessive distortion or rag of piping (e.g. thermo plastics) subject to creep under conditions of repeated thermal cycling.
10. Excessive heat flow, exposing supporting elements to temperature extremes outside their design limits.

ASME B 31.1 deals with the design of supporting elements under clause 121.

It was indicated in Stress Analysis that when the piping is connected to strain sensitive equipments, the allowable forces and moments govern the analysis. In practice the magnitude of these forces and moments is controlled by the use of

- Anchors – controls all 6 degrees of freedom
- Line stops
- Guides – to allow the pipe to move in axial dir
- Rests – to stop the movement in -y dir

We will analyze each type of support as below.

In addition to the weight effects, consideration shall be given for other load effects induced by service pressure, wind, earthquake etc. Where the resonance with imposed vibration and for shock occurs during operation, suitable dampners, restraints, anchors etc shall be added to remove these effects.

Anchors are provided to secure the desired points of piping whereas **guides** are provided to direct or absorb the same. They shall permit the piping to expand and contract freely away from the fixed points. **Sliding** or **Rest** supports permit free movement of piping and shall be designed to include friction resistance along with the dead weight of the piping. **Resilient** supports are those which support the dead weight throughout the expansion / contraction of the piping.

The '**primary support**' is the supporting element, which is attached, or in contact with the piping and the '**secondary support**' is the supplementary steel provided to carry the load on to the structures.

1.1 Anchors

At an anchor, a pipe is assumed to be completely restrained against any displacement or rotation, relative to the structure to which it is attached. This is a point where all the six degrees of freedom are arrested.

It is possible for an anchor to have a displacement or rotation imposed upon it by influences external to the piping system and which it then transmits on to the piping system. For example, consider one end of a pipe, which ends at a nozzle near the 'sliding' end of a shell, and tube heat exchanger. The Piping Engineer would regard that nozzle as "anchor" for the purposes of calculation, but it will move by an amount determined by the thermal expansion of the exchanger shell and it will impose this movement on the piping

Type of structures – Angle Channel I-Beam (If L≠M) Wide Beam (If L=M)

25 800

anchored to it; this movement will then either add or subtract from the restrained thermal expansion of the piping as the case may be.

Anchors may be fitted at points other than the terminations of a pipe in which case they are known as "intermediate anchors" and in this sense the great majority of anchors used in piping installations are of this intermediate category. These anchors serve the purpose of defining fixed points in the system.

As an example of this use of an anchor, consider the case where a 6"(150 mm) NB branch to the inlet of a Turbine taken fro 12" (300 mm) NB header.

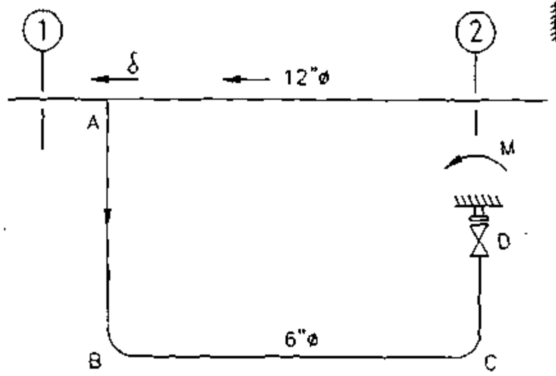


Fig. 1.1

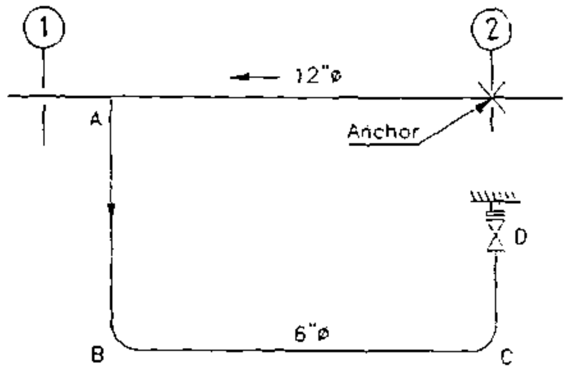


Fig.1.2

Suppose that a large movement δ of the 12 in. (300 mm) header at "A" produced unacceptable forces and moments at the Turbine flange 'D', and that it was not possible to increase the flexibility of the intervening pipe ABCD. We could then try anchoring the header on the piperack beam (2) as shown in Fig 1.2 and making a check on the forces and moments developed by the shape ABCD with only this section of the header included in the calculation.

On very long piperack runs, where more than one expansion loop is required to absorb the expansion between given terminal points, intermediate anchors MUST be fitted between each pair of expansion loops even though the line is of uniform size and the loops are nominally identical, as in Fig. 1.3.

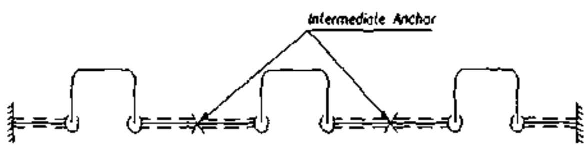


Fig.1.3

The reason for this requirement is that the pipe is subject to manufacturing tolerances in wall thickness so that even though the loops have the same overall dimensions they will have somewhat different flexibilities and in the absence of intermediate anchors which define the amount of expansion taken by each loop, one or other of them would take more than its calculated share of the total movement.

1.2 Line Stops

A line stop is a restraint, which prevents any axial movements of the pipe to which it is fitted but at the same time allows unrestricted travel in any direction at right angles to the axis of the pipe. It also permits rotation, freely, in any plane.

In many instances where the requirement is for an axial restraint only, a "Line stop" can be substituted for a 'full' anchor. Situations do arise where the ability of the 'stop' to permit lateral movement makes its use imperative.

Suppose we have the "header and branch" situations illustrated in Fig. 1.4 – 1.7

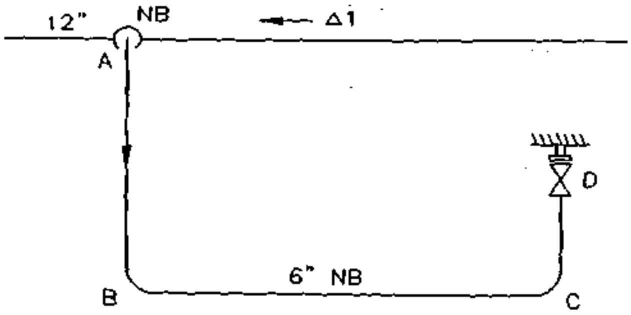


Fig.1.4

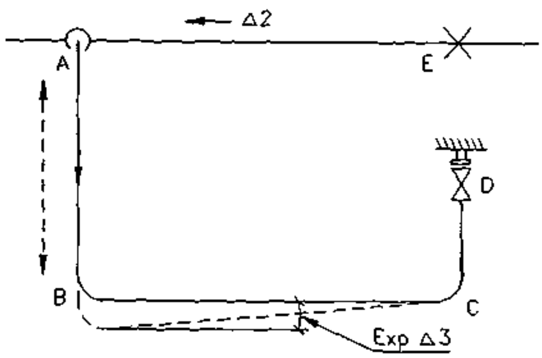


Fig.1.5

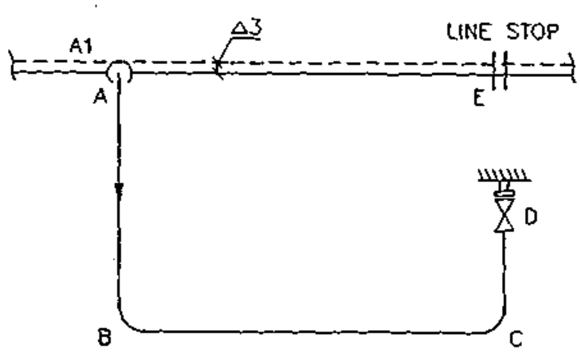


Fig.1.6

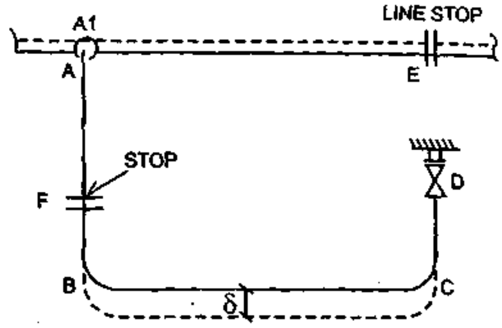


Fig.1.7

In the Fig 1.4 we have a 6"(150 mm) NB. branch ABCD from a 12" (300 mm) NB header, and analysis shows that the forces and moments at 'D' due to the deflection Δ_1 are excessive.

In Fig.1.5, we have added an anchor at 'E' which reduces the axial movement at 'A' to Δ_2 but a check analysis shows that the branch ABCD *is still* overstressed as a result of the restrained expansion Δ_3 due to the length of AB.

The remedy is shown in the Fig.1.6 above, where the anchor at 'E' has been replaced with a 'line stop'. This allows the point 'A' to move over to the position 'A_l' thereby relieving the forces and moments due to the restraining of Δ_3

In the situation shown in Fig.1.6 above, there must be an axial compression force along the leg 'AB' sufficient to move the 12" header sideways against the friction forces exerted at its supports. This force could, in some instances, exceed that which can be taken on the nozzle at 'D'. The problem can be resolved by the addition of a further 'Stop' on the leg 'AB' as shown at 'F' in Fig.1.7.

Ideally, 'F' should be located so that the deflection δ at 'C', due to the nozzle movement plus the thermal expansion of the leg 'DC' is balanced by a corresponding δ

due to the thermal expansion of the length 'FB'. In practice, this ideal situation is difficult to realize but a suitable compromise position can usually be found. This compromise solution could well entail additional structural steelwork.

1.2 Guides

A guide is a restraint, which precludes lateral movement of the pipe in one or both of the planes at right angles to the pipe centerline. It leaves the pipe completely free to move axially and it offers no resistance to rotation of the pipe in any direction.

Guides are provided whenever it is necessary to maintain the position of the centerline of the pipe and some of the more common applications are as follows.

1.3.1 ALIGNMENT GUIDES IN A PIPE RACK

The center to center spacing of the pipes in a rack is such that they must be positively located at intervals along their length.

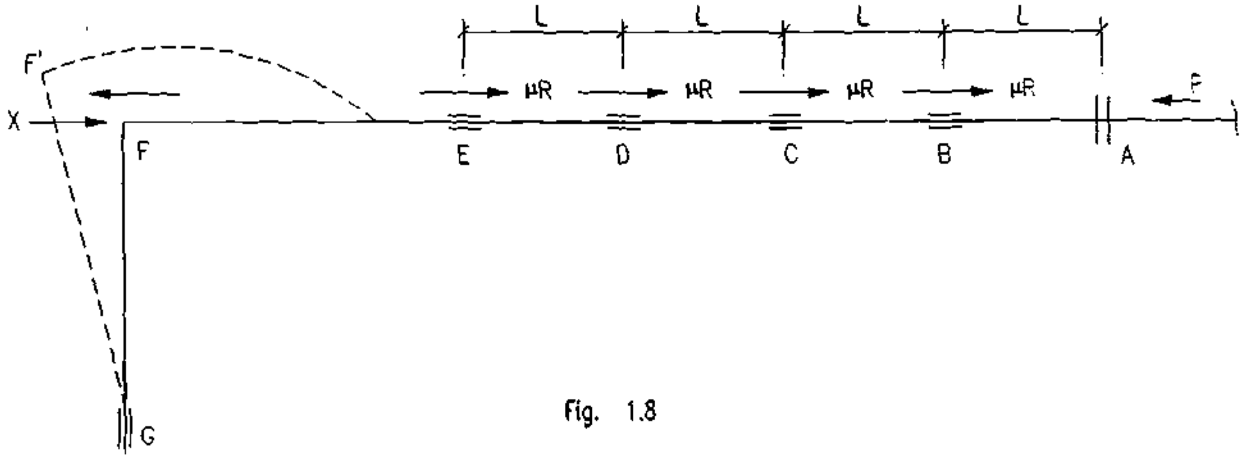


Fig. 1.8

Suppose that we have a length of pipe in a rack as shown in Fig.1.8. There will be an axial force 'X' at F due to the thermal expansion of the length 'AF' and this is balanced by the reaction at the line stop at 'A'. The force 'X' deflects the length 'FG' to the position F'G, and its magnitude can be conservatively estimated by the Guided Cantilever method. Furthermore, at each of the supports, B, C, D and E, there will be a friction force μ R; as the pipe expands and this will add to the basic flexibility force 'X'.

At any position along the pipe we, therefore, have a compression force 'P' given by

$$P = X + \Sigma\mu R$$

It should be noted that this only occurs in practice on small bore pipework.

$$P_{cr} = \pi^2 EI/L^2$$

Where:

P_{cr} is the buckling load in kgs

E is Modulus of Elasticity,

I is Moment of Inertia of pipe cross section .

L is the guide spacing along the rack in meters

Graphite is some times provided under steel pipes to reduce friction as μ in this case μ=0.1

In practice, it would be prudent to limit the practical guide spacing to something of the order of 70% of the value given by the above expression. Also, practically, the length 'L' will be fixed by the overall design of the rack, and since the rack frame spacing will have been settled from other considerations long before any pipe support work is commenced, the location of the position of guides becomes a matter of deciding whether they shall be fitted at every one, two or three frame spacing. Note that because the friction component of the compression forces in the pipe reduces as one gets further from the anchor, it may not be necessary to maintain a uniform guide spacing throughout the full length.

1.3.2 WIND GUIDES ON VERTICAL LINES

A number of lines extended from near Grade elevation to various positions on a fractionating column. Particularly at higher elevations, the wind loading on a pipe can be quite considerable and the line must be guided at intervals.

Load at each grade is given by

Load = wind pressure x projected area of pipe between guides

An average value for the spacing of wind guides on a column is between 8m and 12m depending on diameter. They are commonly designed to restrain both radial and circumferential movement and as such are often referred to as 'boxed' guides.

1.3.3 GUIDES AT PUMP SUCTION NOZZLES

One way of relieving high forces and moments at the suction nozzle of a front suction pump is to fit a guide which is made to very close tolerances and which has an appreciable length. It is argued that this guide is capable of absorbing any terminal moments in addition to the side shear forces. The pump may, therefore, be considered protected from the effects of the piping loads whilst the guide construction permits free movement of the nozzle arising from the expansion of the pump casing as well as the straight length of pipe.

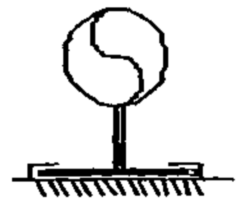


Fig. 1.9

It should be understood, however, that it is customary to provide a plain guide and a support at the suction nozzle of a front entry pump.

1.3.4 GUIDE ON PIPERACK

Use of the Guided Cantilever analysis in deciding the location of guides in a pipe rack is as illustrated below:

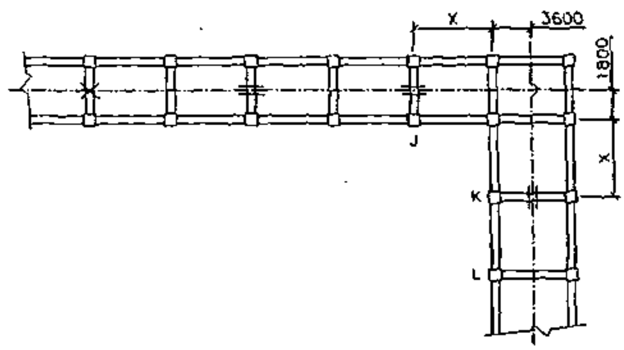


Fig. 1.10

Guide is to be positioned on the piperack in such a way that the force exerted due to thermal expansion should not be excessive for the structural arrangement provided.

Clearly, if the guide which was positioned on frame 'K' had instead been located on Column Row 'J', the sideways force on it would have been given by;

$$P = EI.\Delta\ /144.L^3$$

Where 'L' now had the value of X + 1.8m instead of X + 3.6m. The value of the force gets multiplied in the 3rd degree of this factor.

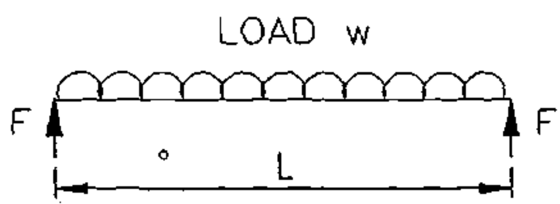
The practical implication of this is that the guide would have broken long before this sideways load had developed, possibly causing a permanent set in the pipe during the process; This situation is now known in practice, hence the reason for the warning in the Code.

2.0 THE DETERMINATION OF SUPPORT LOCATIONS

The support location is dependent on the pipe size, piping configuration, the location of heavy valves and specialties and the structure available for support. The simplest method of estimating the support load and pipe stress due to weight is to model the pipe as a beam loaded uniformly along the length, the length of the beam equal to distance between supports.

There are two possible ways to model the pipe, depending upon the end conditions – the simply supported (pinned end) beam or the fixed end beam.

For a simply supported beam, the maximum stress and support loads are.



$$M_{max} = \frac{wl^2}{8}$$

$$f = \frac{wl^2}{8Z}$$

$$F = \frac{wl}{2}$$

where,

M_{max} = maximum bending moment, ftlb (Nm)

f = Bending stress, psi (N/mm^2)

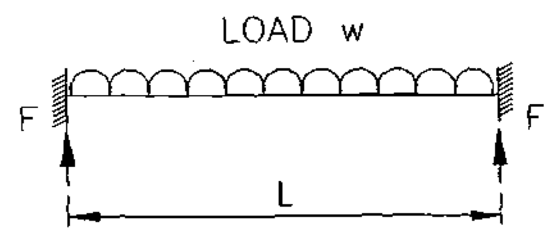
w = weight per unit length, lb/in (N/mm)

l = length of pipe, in (mm)

F = force on support, lb (N)

Z = section modulus in^3 (mm^3)

For fixed end beam



$$M_{max} = \frac{wl^2}{12}$$

$$f = \frac{wl^2}{12\,Z}$$

$$F = \frac{wl}{2}$$

For either model, the support load remains the same. However, depending upon the model chosen the stress in pipe varies. In actual practice the pipe at the point of support is not free to support fully, since it is partially restrained through its attachment to piping segment beyond the support. If the pipe runs between supports are equally loaded and of equal length, segment end rotation could cancel each

other causing the pipe to behave as fixed-end beam. Therefore, the true case lies somewhere between the two beam models. Hence, as a compromise case, the stress is calculated as

$$f_{max} = \frac{wl^2}{10\,Z}$$

Hence, support spacing is decided by the formula

$$l = \sqrt{\frac{10\,Z\,S}{w}}$$

where
S is the allowable stress as per the code in psi (N/mm^2)

The suggested maximum spans between the supports, as recommended by ASME B 31.1 in Table 121.5, are as follows. This is adapted by ASME from the MSS-SP standard MSS-SP-69

Nominal Pipe Size, NB Inch	Suggested Maximum Span: Water Service m (ft)	Steam Gas Air Service, m (ft)
1	2.1(7)	2.7(9)
2	3.0(10)	4.0(13)
3	3.7(12)	4.6(15)
4	4.3 (14)	5.2(17)
6	5.2(17)	6.4(21)
8	5.8 (19)	7.3 (24)
12	7.0(23)	9.1(30)
16	8.2(27)	10.7 (35)
20	9.1(30)	11.9(39)
24	9.8 (32)	12.8 (42)

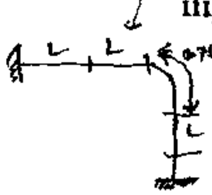
The above spacing is based on fixed beam support with a bending stress not to exceed 2300 psi and insulated pipe filled with water or the equivalent weight of steel pipe for steam, gas or air service and 2.5mm (0.1 inch) sag is permitted between supports. The suggested maximum spacing between supports is for horizontal straight runs of standard and heavier pipes at a maximum operating temperature of 400^{0}C (750^{0}F).

These do not apply where there are concentrated loads between supports such as flange, valve, specialties etc. and also where change in direction occurs between supports.

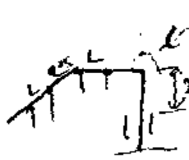
The location of supports should consider the following guidelines:

i) The support should be located as near as possible to concentrated loads such as valves, flanges etc. to keep the bending stress to the minimum.

ii) When changes of direction in a horizontal plane occur, it is suggested that the spacing be limited to 75% of the tabulated values to promote stability and reduce eccentric loadings. Note that the supports located directly on elbows are not recommended since that will stiffen the elbow and no flexibility will be available.



iii) The standard span does not apply to vertical run pipes (risers) since no moment and no stress will develop due to gravity load in the riser. The support should be located on the upper half of a riser (above the center of gravity) to prevent instability in overturning of pipe under its own weight. Guides may be placed on long vertical risers to reduce pipe sag resulting in excessive pipe deflection. These guides are usually placed in span intervals of twice the normal horizontal span and do not carry any dead weight.



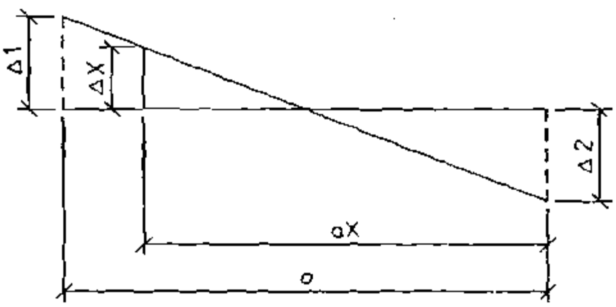
support span in ft = Pipe size in (inch) + 10 (above 4")

iv) Support location should be selected near the existing building steel to minimize the use of supplementary steel.

In case of pipeline running in multiplane, the support load is determined by applying a method called 'weight balancing'. This method involves breaking the larger piping system into smaller segments of pipe with supports, which are modeled as free bodies in equilibrium and solved statically.

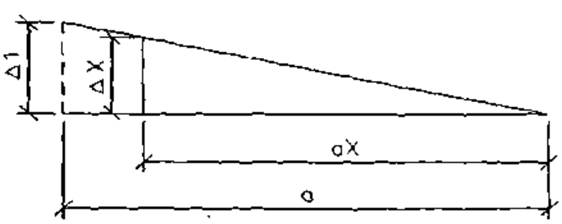
3.0 DISTRIBUTION OF MOVEMENTS

The movement at the specified location is calculated based on the known movements using the following methods.



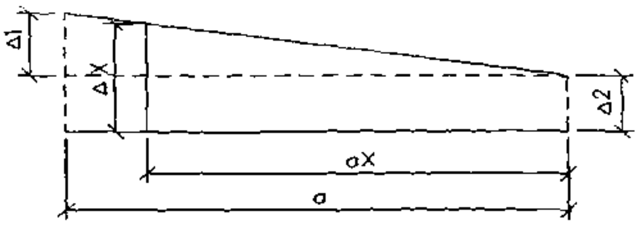
$$\Delta X = \frac{aX(\Delta 1 + \Delta 2) - a\Delta 2}{a}$$

CASE- 1



$$\Delta X = \frac{aX(\Delta 1)}{a}$$

CASE- 2



$$\Delta X = \frac{aX(\Delta 1)}{a} + \Delta 2$$

CASE- 3

4.0 PROBLEM APPROACH

Let us apply these principles to a simple practical problem.

In the illustrated example seven supports are shown on a 6"(150 mm) NB pipeline routed to join the equipment nozzle 'A' with the equipment nozzle 'B'.

4.1 Location of Supports

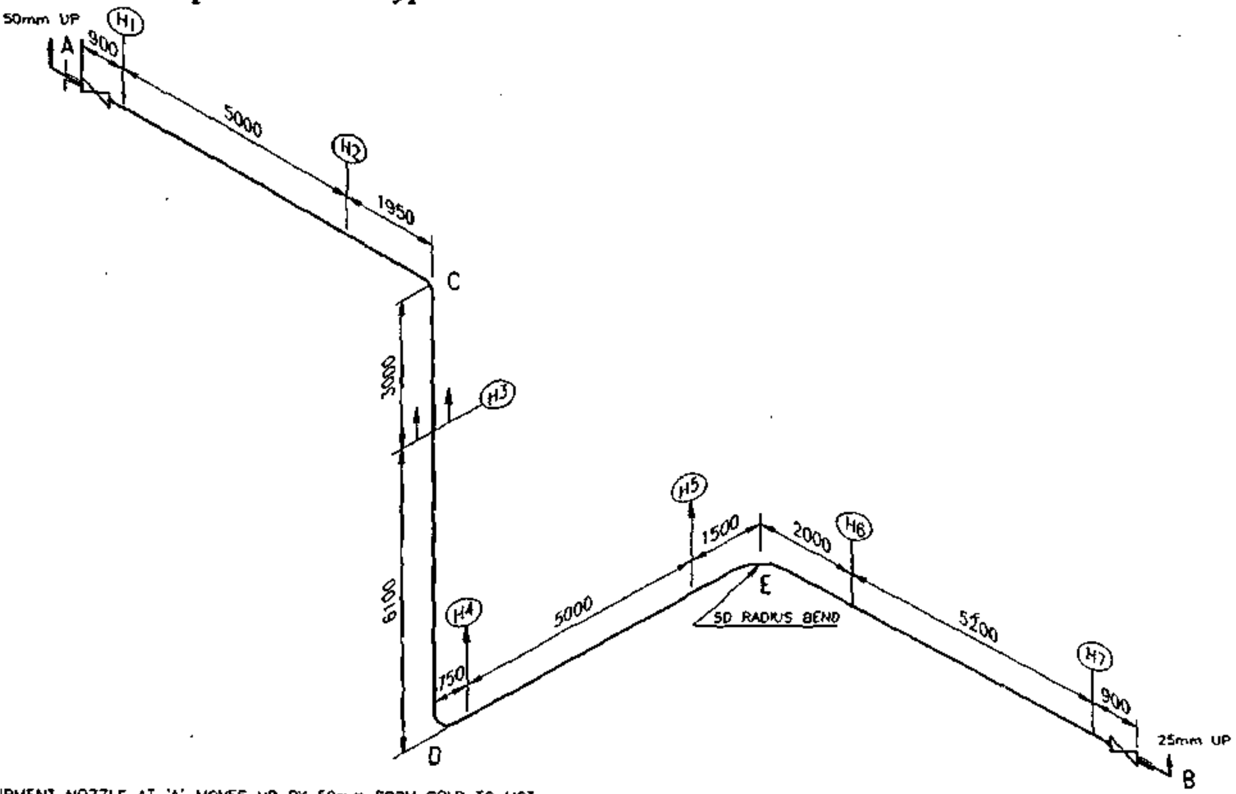
The hanger H1 is placed adjacent to the valve to keep the load at the equipment to the minimum. Location of H2 is governed by the suggested maximum span. Support H3 on vertical leg is placed above the center of gravity. Calculations would indicate that the CG of the vertical leg falls at about 5000 mm above the lower horizontal run. If the support is placed below the center of gravity, an unstable turnover condition would result. The support would then act as a pivot and would not resist sway. It could also be checked with minimum leg calculation

The location of H7 is kept adjacent to the valve weight concentration. The proximity of the support to the valve is helpful in keeping the load at the equipment flange B to the minimum or nil as required. The hanger H6 is located considering the suggested maximum span.

The selection of location for H5 entails a change in direction of pipe between two hangers. In order to avoid excessive overhang of the pipe between hangers H5 and H6, the developed length

of pipe between the hangers is kept less than three fourth (3/4) of the suggested maximum span. The hanger H4 is located clearing the bend at point D and within the maximum suggested span from H5.

The method involved in locating hangers for this problem are typical of those employed by Piping Engineer in the design of support. Although the individual piping configurations and structure layout will vary practically in every instance; the general methods outlined above will apply for any critical piping system.



NOTES ::

EQUIPMENT NOZZLE AT 'A' MOVES UP BY 50mm FROM COLD TO HOT.
EQUIPMENT NOZZLE AT 'B' MOVES UP BY 25mm FROM COLD TO HOT.
PIPE– 150 NB SCH 160 ASTM A335 Gr P12.
OPERATING TEMP. 550° C
ALL ELBOWS ARE L R ELBOWS.

ILLUSTRATED EXAMPLE - PIPE HANGER DESIGN

4.2 The Determination of Thermal movement of the Piping at each Support Location.

The next step in the design of pipe hangers involves the calculation of the thermal movement of the pipe at each hanger location. The simplified method illustrated here is the one, which gives satisfactory approximation of piping movements. However this approximation will always give positive error. The distribution of the movement at the various supporting points are done based on the following arrangement.

Step I

Draw the piping system under analysis as shown and indicate all terminal movements from its cold to operating position. Those movements will include those supplied by equipment manufacturers for the terminal connections.

For the illustrated problem the following vertical movements are known,
Point A - 50 mm up, Cold to Hot
Point B - 25 mm up, Cold to Hot
The above data is as furnished by the manufacturers of equipment.

H3 - 0 mm Cold to Hot

The pipeline is 6 Inch (150 mm) NB Alloy Carbon Steel pipe carrying high-pressure steam. The operating temperature of the system will be 550^0C. Referring to 'Thermal Expansion Data' Appendix B, ASME B 31.1 (or Appendix C 1 ASME B31.3), the Coefficient of thermal expansion for low chrome steel at 550^0C (1022^0F) is 0.09143 inch/ft. i. e. 7.62 mm/m

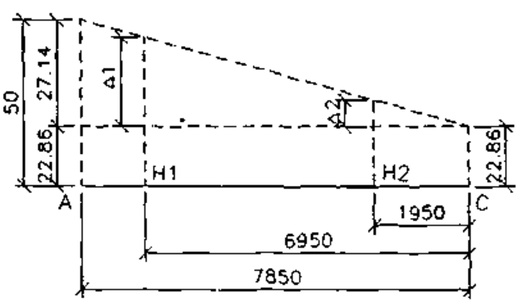
Calculate the expansion at point C and D by multiplying the Coefficient of expansion by the vertical distance of each point from the position of zero movement on the riser CD.

3.0x7.62 = 22.86 mm up at point C

6.1 x 7.62 = 46.48 mm down at point D

Step II

Make a simple sketch between two adjacent points of known movement



(Refer Case 3 of "Distribution of movements")

The vertical movement at hanger location can be calculated by proportioning the same.

$$\Delta 1 = \frac{6950 \times 27.14}{7850} = 24.03$$

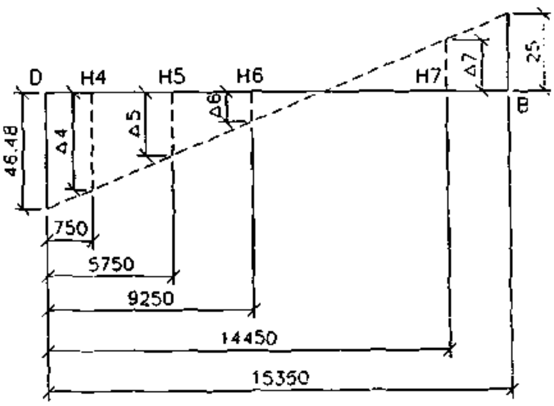
Vertical movement of Δ H1 = 22.86+24.03
= 46.89
Say 47 mm
i.e. **47 mm up**

$$\Delta 2 = \frac{1950 \times 27.14}{7850} = 6.74 \text{ mm}$$

Vertical movement at H2
= 22.86+6.74
= 29.60 **Say 30 mm**
i.e. **30 mm up**

Step III

Make the sketch of piping between the points B and D, extending the piping to a single plane as shown.



(Refer Case 1 of "Distribution of movements")

$$\Delta 4 = \frac{750(46.48 + 25) - 15350 \times 46.48}{15350}$$

= -42.99 mm say **-43 mm**

Vertical movement at H4 = **43 mm down**

$$\Delta 5 = \frac{5750(46.48 + 25) - 15350 \times 46.48}{15350}$$

= -19.70 mm say **-20 mm**

Vertical movement at H5 = **20 mm down**

$$\Delta 6 = \frac{9250(46.48 + 25) - 15350 \times 46.48}{15350}$$

$= -3.41$ mm say **-3 mm**

Vertical movement at H6 = **3 mm down**

$$\Delta 7 = \frac{14450\,(46.48 + 25) - 15350 \text{ X } 46.48}{15350}$$

$= 20.81$ **say 21 mm**

Vertical movement at H7 = **21 mm up**

For easy reference when selecting the appropriate hanger, let us make a simple table of hanger movement

Hanger Number	Movement (mm)
H1	47 up
H2	30 up
H3	0
H4	43 down
H5	20 down
H6	3 down
H7	21up

4.3 The Calculation of load at each Support Location

The thermal expansion of high temperature piping makes it necessary for the Piping Engineer to specify flexible support, thereby requiring considerable thought to calculation of hanger loads. High degree of accuracy is required in determining the support force, in order to select the appropriate type and size of spring support.

The calculation of the loads for hangers involves dividing the system into convenient sections. A free body diagram of each section should be drawn to facilitate the calculation with simple arithmatic solution to the problem.

The first step in the solution is to prepare a table of weights. For the illustrated problem, the table shall be as below

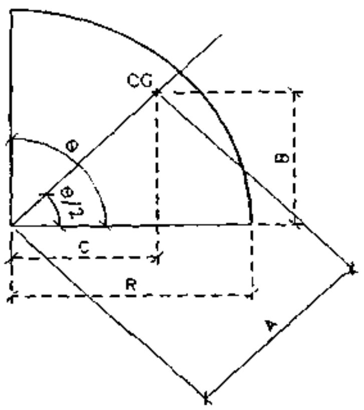
Table of Weights

Description	Weight	Weight of Insln (CaSi)	Total Weight	Weight used in Calculation
150 NB Sch 160 pipe	67.5 kg/m	17.0 kg/m	84.5 kg/m	84.5 kg/m
150 NB Sch 160 90° BW LR Elbow	24.0 kg	8.0 kg	32 kg	32 kg
150 NB BW 1500 lb class Gate Valve	725 kg	37.0 kg	762 kg	762 kg

Draw a free body diagram of the piping between point A and H1 showing all supporting and all valve and pipe weights. The load at the supporting point can be calculated as indicated in Fig.4.1. Note that the value of H1 on this section of piping system represents only part of the total load at H1. This has to be combined with the load available by the calculation of the next section starting at H1 as indicated in Fig.4.2

Fig.4.3 is the free body diagram of the section of piping between H2 and H3. It is considered that the weight of 90^0 bend acts at the center of gravity of the bend.

The various distances to the center of gravity of the bend can be calculated using the formula as below:



$$A = \frac{2R \sin \theta/2}{\theta}$$

$$B = \frac{R\ (1 - \cos \theta)}{\theta}$$

$$C = \frac{R \sin \theta}{\theta}$$

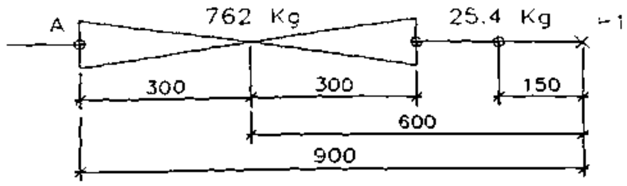
Applying the above formula for the distance of CG from the center of the arc for 150 NB LR elbow.

$$C = \frac{R \sin \theta}{\theta}$$

$$= \frac{229.0 \times 1}{\pi / 2}$$

$$= 145.8\text{mm}$$

Distance of the CG form the center line of the straight pipe = 229.0 - 145.8 = 83.2 mm



Taking moments about H1,

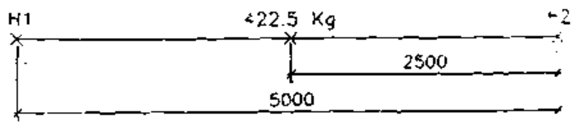
m	x	kg.	=	kg.m
0.15	x	25.4	=	3.81
0.60	x	762.0	=	457.20
		787.4		461.01

$$\text{Reaction at the point A} = \frac{461.01}{0.9}$$

$$= 512.23\text{kg}$$

Reaction at the point H1 = 787.4 - 512.2 = 275.17 kg.

Fig. 4.1 Distribution Of Load Between Equipment Connection A & H1



$$\text{Reaction at the point H1 \& H2} = \frac{422.5}{2}$$

$$= 211.25\text{ kg}$$

Fig. 4.2 Distribution Of Load Between H1 & H2

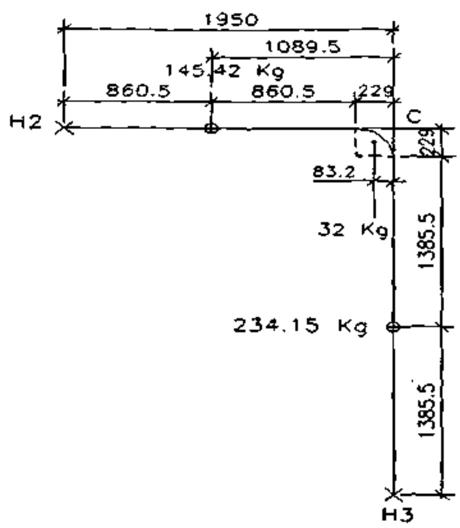


Fig.4.3 Distribution Of Load Between H2 & H3

Taking moments about H3

m	x	kg.	=	kg.m
0	x	234.15	=	0
0.0832	x	32.00	=	2.66
1.0895	x	145.42	=	58.44
		411.57		161.10

$$\text{Reaction at H2} = \frac{161.10}{1.95}$$

= 82.62 kg

Reaction at H3 = 411.57 - 82.62

= 328.95 kg.

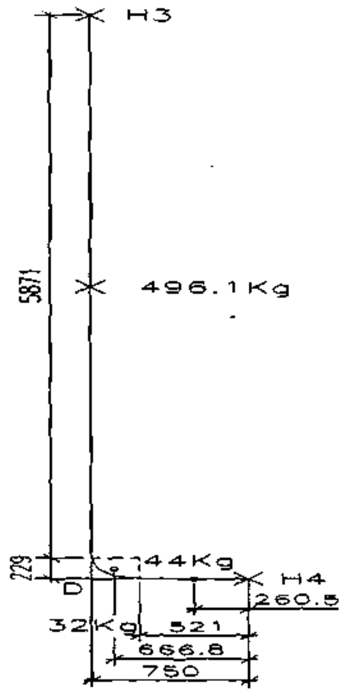


Fig. 4.4 Distribution of Load Between H3 & H4

Taking moments about H4

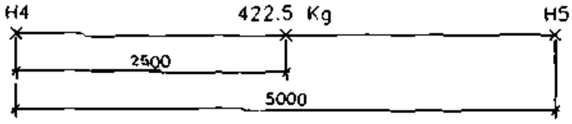
m	x	kg.	=	kg.m
0.2605	x	44.0	=	11.46
0.6668	x	32.0	=	21.34
0.750	x	496.1	=	372.08
		572.1		404.88

$$\text{Reaction at H3} = \frac{404.88}{0.750}$$

= 539.84 kg

Reaction at H4 = 572.1 - 539.84

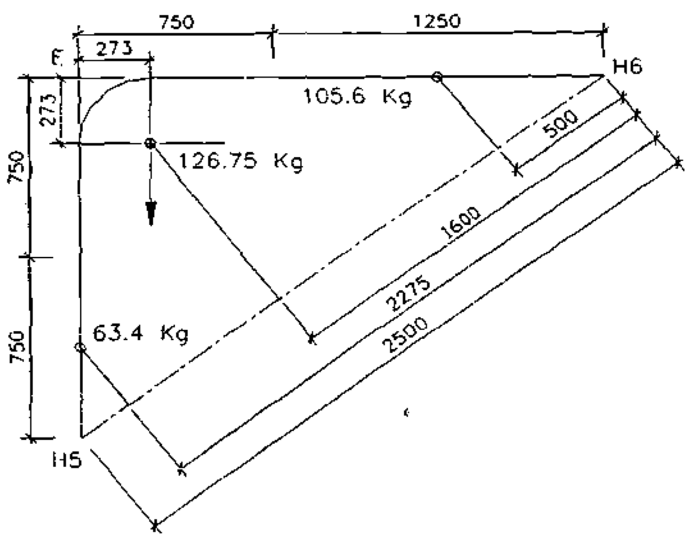
= 32.26 kg.



$$\text{Reaction at the point H4 \& H5} = \frac{422.5}{2}$$

$$= 211.25 \text{ kg.}$$

Fig. 4.5 Distribution Of Load Between H4 & H5



Taking moment about H6

m	x	kg.	=	kg.m
0.5	x	105.6	=	52.8
1.60	x	126.75	=	202.80
2.275	x	63.4	=	144.2
		295.75		399.8

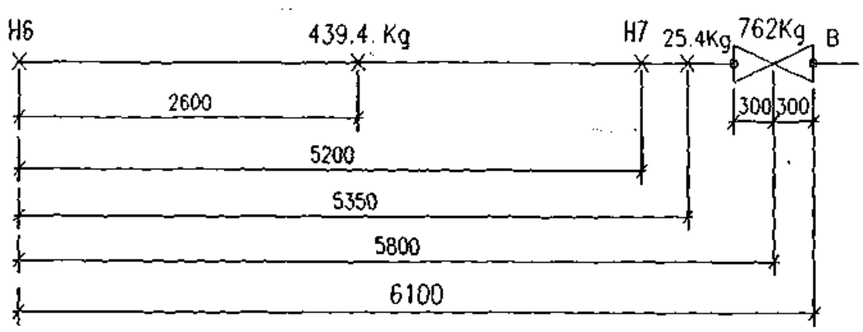
$$\text{Reaction at H5} = \frac{399.8}{2.5}$$

$$= 159.92 \text{ kg}$$

$$\text{Reaction at H6} = 295.75 - 159.92$$

$$= 135.83 \text{ kg}$$

Fig. 4.6 Distribution Of Load Between H5 & H6



Taking moment about H6

m	x	kg	=	kg.m
2.60	x	439.4	=	1142.44
5.35	x	25.4	=	135.89
5.80	x	762.0	=	4419.60
		1226.8		5697.93

As the nozzle B is relieved of load

$$\text{Reaction at H7} = \frac{5697.93}{5.2}$$

$$= 1095.76 \text{ kg}$$

$$\text{Reaction at H6} = 1226.8 - 1095.76$$

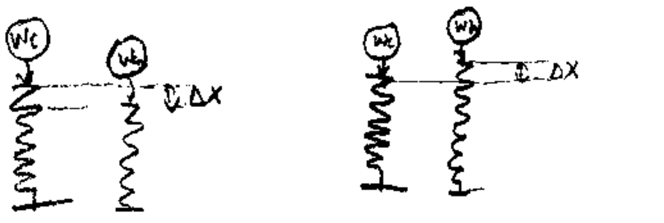
$$= 131.04 \text{ kg.}$$

Fig. 4.7 Distribution Of Load Between H6 & H7 To Maintain Zero Reaction At Nozzle B

Hot load - Operating load

Cold load - Install load

$$\frac{\text{operating load} - \text{inst load}}{\text{operating load}} = \text{\% Load variation}$$



(In case spring compression)

$W_c > W_h$

$W_c = W_h + K\Delta x$

(In case of spring expansion)

$W_c < W_h$

$W_h = W_c + K\Delta x$

SUMMARY OF LOADING

LOCATION	REACTION FOR LOADING							HANGER LOAD KG
	A TO H1	H1 TO H2	H2 TO H3	H3 TO H4	H4 TO H5	H5 TO H6	H6 TO H7	
NOZZLE A	512,23							512,23
H1	275.17	211.25						486.42
H2		211.25	82.62					293 87
H3			328,95	539,84				868.79
H4				32,26	211,25			243,51
H5					211.25	159.92		371 17
H6						135.83	131.04	266.87
H7							1095,76	1095.76
8							0.00	0,00
TOTAL								4138.62

WEIGHT OF PIPING SYSTEM

150NB PIPE @ 84,5 Kg/M	28,684 x 84.5	2423,80
150NB BW 1500lb CLASS GATE VALVE @ 762 Kg	2 x 762	1524,00
150NB LR BW ELBOW @ 32 Kg	2 x 32	64,00
150NB 5D BEND @ 126.75 Kg	1 x 126 75	126.75
TOTAL		4138.55

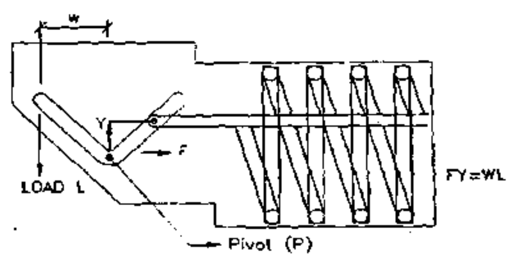
5.0 THE SELECTION OF FLEXIBLE SUPPORTS

When vertical displacement occurs as a result of thermal expansion, it is necessary to provide a flexible support, which applies supporting force throughout the contraction and expansion cycle of the system

Flexible hangers are of two types: Constant Spring and Variable Spring

5.1 Constant Spring

Constant spring hangers are selected where absolutely necessary, when the percentage variation of load from cold to hot should be less than ± 6% for critical pipelines and less than ± 25% for non critical pipe lines and when it cannot be obtained by the type of variable spring hangers, which give the lowest %, load variation. The geometry and kinematics of these constant spring hangers is such that theoretically a constant supporting force can be achieved throughout its full range of expansion and contraction. This is accomplished by the use of a helical coil spring working in conjunction with a bell crank lever in such a way that the spring force times its distance to the lever pivot is always equal to the pipe load times its distance to the lever pivot.



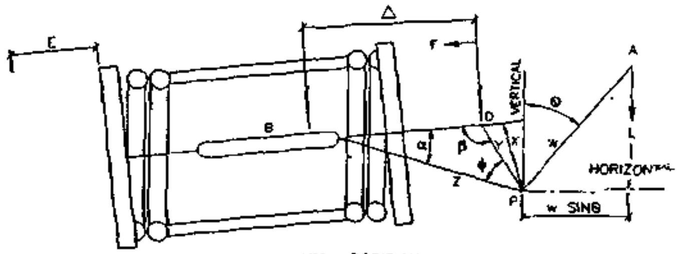
(FY = WL)

This counter balancing of the load and spring movements about the main pivot is obtained by the use of carefully designed compression type load springs, lever and spring tension rods.

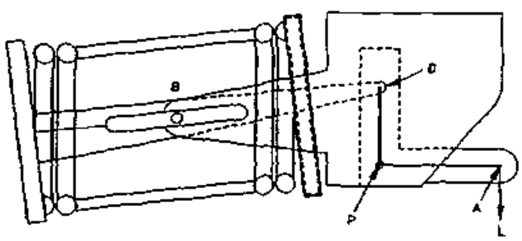
As the lever moves from the high to low position, the load spring is compressed and the resulting increasing force acting on the decreasing spring moment arm creates a turning moment about the main pivot which

is exactly equal and opposite to the turning moment of the load and load moment arm.

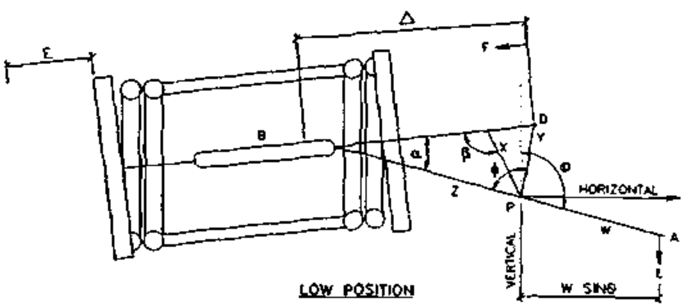
As the lever moves from the low to high position, the load spring is relaxed and the resulting decreasing force acting on the increasing spring moment arm creates a turning moment about the main pivot which is exactly equal and opposite to the turning moment of the load and load moment arm.



HIGH POSITION



MID POSITION



LOW POSITION

$$\frac{Y}{\sin\alpha} = \frac{\Delta}{\sin\phi} = \frac{Z}{\sin\beta}$$

Considering, $\frac{Y}{\sin\alpha} = \frac{Z}{\sin\beta}$

$$\sin\alpha = \frac{Y\sin\beta}{Z}$$

Since $Y \sin\beta = X$

$$\therefore \sin\alpha = \frac{X}{Z}$$

Substituting in Eqn. $\frac{Y}{\sin\alpha} = \frac{\Delta}{\sin\phi}$

i.e. $\frac{Y}{X/Z} = \frac{\Delta}{\sin\phi}$

$$\frac{YZ}{X} = \frac{\Delta}{\sin\phi}$$

or $X = \frac{YZ\sin\phi}{\Delta}$

The Load 'L' is suspended from the lever at point 'A' and at any point within the load travel range the moment of the load about the main lever - pivot 'P' is equal to the load times its moment arm.

Thus Load moment = L (W Sinθ), where W Sinθ is the load moment arm.

The spring is attached to one of its ends to the fixed pivot "B". The free end of the spring is attached by means of a rod to the lever-pivot 'D". This spring arrangement provides a spring moment about the main lever-pivot "P" which opposes the load moment and is equal to the spring force, "F" times its moment arm.

Thus spring moment = $FX = \frac{F(YZ\sin\phi)}{\Delta}$

Where X is the spring moment arm

The spring force "F" is equal to the spring constant "K" times to the spring deflection "E"

Thus $F = KE$

$$\text{Spring Moment} = \frac{KE\,(Y\,Z\,\sin\phi)}{\Delta}$$

To obtain PERFECT constant spring, the load moment must always equal to spring moment.

$$LW \sin\theta = \frac{KEYZ\,\sin\phi}{\Delta}$$

By proper design ϕ and θ are made equal

$$\text{Therefore } LW = \frac{KEYZ}{\Delta}$$

The spring and the rod are so designed that the spring deflection "E" always equals the distance "Δ" between pivots "B" and "D"

$$\text{Therefore } LW = KYZ$$

$$\text{or} \quad L = \frac{KYZ}{W}$$

This equation holds true for all position of load within its travel range and "K", "Y", "Z" and "W" remain constant. It is therefore true that perfect constant support is obtained.

But due to spring hysteresis, bearing friction, sliding friction of moving parts and manufacturing tolerances, it is not normally possible to keep constant effort throughout the travel range. The deviation is kept very minimum by using PTFE washers and bushes at all pivot points and lifetime lubricated antifriction bearings.

There are different models of constant springs available based on the type of supporting arrangement. These are manufacturer specific and generally as below.

a) Spring located horizontally with the supporting structure above and the supported pipeline below the spring called model "H" by the manufacturers.

b) Spring located horizontally with the supporting structure below and the supported pipeline also below the spring called model "E" by M/s Sarathy and model "M" by M/s Myricks.

c) Spring located horizontally with the supporting structure below and the supported pipeline above the spring called model "F" by M/s Sarathy and model "S" by M/s Myricks.

d) Spring located vertically with the supported structure above and the supported pipeline below the spring called "V" by the manufacturers.

e) Spring located vertically with supporting structure below and the supported pipeline above the spring called model "P" by M/s Myricks.

5.1.1 HOW TO SELECT A CONSTANT SPRING SUPPORT

i) First select the basic model best suited for piping layout and the physical structure available for mounting.

ii) Establish the total travel by giving a positive allowance of about 20% to the calculated actual travel and in no case less than 25 mm in order to allow for a possible discrepancy between calculated and actual piping movement
i.e. Total travel = Actual travel + Over travel

iii) Use the selection table supplied by manufacturer and locate the total travel required at the corresponding table.

iv) Move along the line until load nearest to the operating load to be supported is located such that the load fits in within a reserve range of ± 10% of the average of

the maximum and minimum loads specified.

v) If the total travel lies between the two indicated figures, the loads between the successive travels can be interpolated.

vi) The corresponding hanger size can be read from the respective column.

5.1.2 SPECIFICATION FOR ORDER

The following data is required to be specified while inquiring / ordering for a constant spring.

i) The exact Hot or Operating load required to be supported during the working condition.

ii) Hydrostatic test load.

iii) The total travel (vertical travel)

iv) The direction of travel either upwards or downwards from the erected position.

v) The set pin locking position (Top, Middle, Bottom or as required)

vi) The basic model.

vii) Requirement of bottom accessory components such as rods, clamps etc.

viii) Any hazardous environmental conditions

ix) Any special finish on the body such as galvanizing etc.

x) Tag or Identification number

5.2 Variable Spring

Variable spring hangers are recommended for general use on non-critical piping systems and where constant supports are not required. The inherent characteristic of a variable spring is such that its supporting force varies with spring deflection and spring scale. The vertical expansion of piping causes a corresponding compression or extension of spring and causes a change in the actual supporting effect of the hanger The variation in supporting force is equal to the product of the amount of vertical expansion and the spring scale. Since the pipe weight is the same during any condition, cold or operating, the variation in supporting force results in additional stresses in the piping system. Accepted practice is to limit the amount of supporting force variation to ±25%.

The Spring hangers are specified by the **Series**, the **Type** and the **Hanger size.**

5.2.1 HOW TO SELECT THE SERIES

The selection of the hanger series shall be done to limit the supporting force within the allowable range. In choosing between the series VS1, VS2, and VS3, it must be ensured that the calculated movement will fall within the working load range. The series VS1 has the maximum variation in supporting force and hence is not a competitive selection but an invention of necessity where headroom is not sufficient to use VS2. Good engineering sense combined with available space and reasonable economic considerations should ultimately determine which series of variable spring hangers should be used.

5.2.2 HOW TO DETERMINE TYPE

The type of variable spring hanger to be used depends upon the physical characteristics required by the suspension problem i.e. available headroom; pipe to be supported above the spring or below the spring etc. The type should be selected from the seven standard types available. These are

VS_1, VS_2, VS_3

$VS_1 < VS_2 < VS_3$

Common at calls

$VS1 > VS_2 > VS_3$

identified by Type A to Type G as illustrated in the Fig. 5.1

5.2.3 HOW TO DETERMINE SIZE

For determining the size of the hanger the load deflection table shall be referred. In order to choose the proper hanger size the data required is the actual load or the working load (also called the hot load) and the magnitude and direction of the pipeline movement from cold to hot.

Locate the hot load in the table. To determine the cold load, read the spring scale up or down for the amount of expected movement. *The chart must be read opposite from the direction of pipe movement.* The load arrived is cold load.

If the cold load falls outside the working load range of hanger selected, relocate the hot load to the adjacent column and find the cold load. When both the hot and cold loads are within the working range of a hanger, the size of the hanger is the number found at the top of the column.

Should it be impossible to select a hanger in any series such that both loads fall within the working range, consideration should be given for a constant spring hanger. Once selected, the percentage load variation shall be checked as follows:

$$\text{Load Variation \%} = \frac{\text{Travel x Spring Rate x 100}}{\text{Hot load}}$$

5.2.4 Specification For Order

The following data is required to be specified while inquiring/ ordering for a variable spring

i) The exact Hot or Operating load required to be supported during the working condition.

ii) Hydrostatic test load.

iii) The calculated vertical movement.

iv) The direction of travel either upwards or downwards from the erected position.

v) The hanger series, type and size

vi) The allowable percentage variation of load from cold to hot

vii) Requirement of accessory components such as rods, clamps etc.

viii) Any hazardous environmental conditions

ix) Any special finish on the body such as galvanizing etc.

x) Tag or Identification number.

5.2.5 COMMISSIONING OF SPRING SUPPORTS

i) Securely attach the spring to the building structure by identifying and locating at each support point in accordance with hanger installation drawing. The location should be such that the hanger should be perpendicular in the hot or operating *position / the load should act vertical.*

ii) Make sure the moving parts are unobstructed.

iii) The locking should not be disturbed till complete erection is over. This lock makes the support work as a rigid support during erection, hydrostatic testing or chemical clearing etc.

iv) The locking pins must be removed after the hanger is fully loaded to put

the piping systems into operation. In case of top mounted support, this lock shall be freely removed by the hand after adjusting the distance between the hangers and pipe by rotating the turnbuckle. In case of foot mounted supports the load flange is rotated till it touches equipment / pipe being supported. Then the threaded bush with hexagonal sides is rotated so that it moves up and the load is gradually transferred on to the support. The preset pin becomes loose when the pipe load becomes the preset or factory calibrated load. The support is then ready for use.

v) Once the preset pin is removed the support allows movement up or down by the specified amount of travel in accordance with the expected pipe movement.

vi) When the line is in operation, carefully check the support for its free movement. Generally no further adjustment is necessary. In case of any adjustment, the same shall be achieved by turning the threaded bush with hexagonal sides in case of foot-mounted support or the turnbuckle in case of top mounted support.

5.3 Snubbers

Rigid restraints are usually necessary when the pipe is strong to survive loads such as earthquake or high winds or other dynamic loads such as fluid hammer. But when these restraints are used in high temperature piping at some location it may develop elevated stress levels. In these cases snubbers are used. Snubbers resemble rigid struts, except that they contain a mechanism, which permits movement in presence of static or slowly applied loads; but which locks up during rapidly applied loads. Snubber will resist dynamic loads while permitting the natural and slower thermal growth of pipe a snubber will not act as a weight support. When both weight and vertical dynamic restraint are required at a point of large vertical movement, both spring and snubber are required. The spring will carry weight load while the snubber will restrain dynamic loads, with both supports permitting the required thermal movements. There are two types of snubbers available, hydraulic and mechanical. The hydraulic snubber is made up of a piston and a double chamber filled with viscous fluid. The sealing of this fluid is the main problem associated with it. The mechanical snubbers operate purely on mechanical basis. Mechanical snubbers are also prone to inadvertent locking giving rise to additional pipe stresses.

5.4 Sway Braces

Sway braces are used to limit the effect of pipe vibration. These are little more than variable springs acting in horizontal plane. When sway brace is installed, the spring preload is adjusted to be zero when pipe is in the operating position. Sway braces, like variable springs, do add some expansion stresses in the pipe.

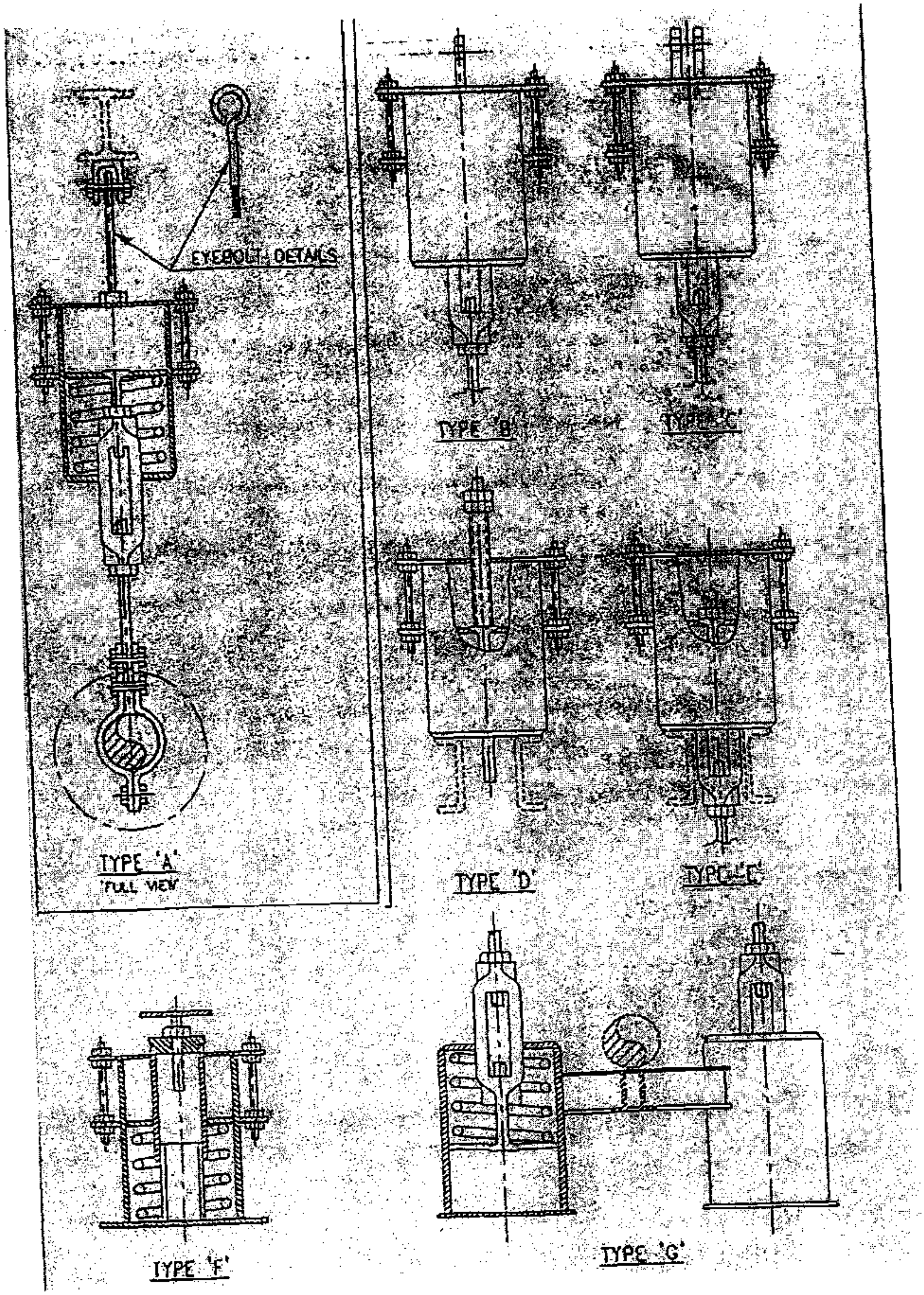


Fig.5.1 TYPE OF SPRING SUPPORTS

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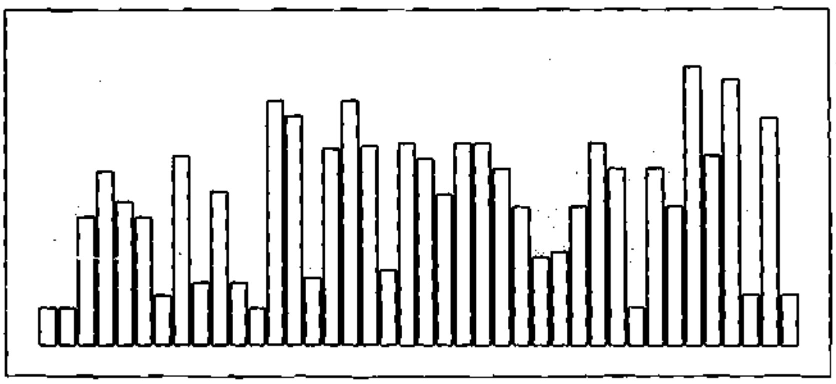
PIPING ENGINEERING

June 12 - 25, 2006

EXPANSION JOINTS

T. N. Gopinath
Consultant



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EXPANSION JOINTS

T. N. GOPINATH

1. INTRODUCTION

When piping lacks inherent flexibility due to routing and/or develops large reactions or detrimental overstrain on the strain sensitive equipments, the Piping Engineer considers provision of expansion joints to overcome the same. Expansion joints are also provided to isolate the vibrating equipment from piping and also to facilitate free movement of the equipment mounted on load cells.

2. TYPES OF EXPANSION JOINTS

The expansion joints can be slip type or the bellows type.

2.1 Slip Type of Expansion Joint

In slip type of expansion joint one pipe slides into another and the assembly is sealed by means of packing between the sliding pipes. This device has the limitation that it permits only axial movement in the direction of pipe axis. Small amount of lateral and/ or angular movement will cause binding and eventually leakage. It is extremely difficult to seal it off completely. The limitations on packing makes it suitable only for very low temperature and low pressure services. Fig 2.1 indicates the general arrangement of a slip type expansion joint.

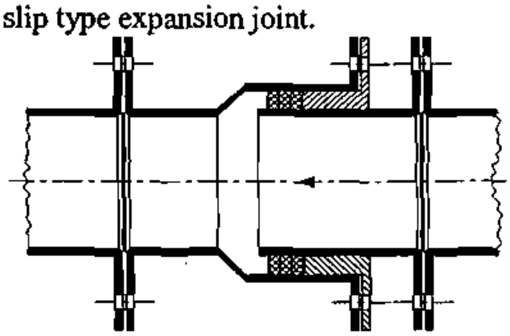


Fig 2.1

2.2 Bellow Type Expansion Joint

The bellow type expansion joint is extensively used as the most efficient and functionally reliable elongation compensator and/ or vibration isolator. These are capable of compensating for large amounts of axial and/ or lateral and/ or angular movements as a single unit. It lends itself to piping configurations that are much more compact than those using bends or loops to provide flexibility.

3. USAGE AND RESTRICTIONS

Based on the above, the point of usage could be identified as below.

- At the suction and discharge nozzles of vibrating equipments such as pumps, blowers etc.,
- On large diameter pipes and ducts operating at high temperatures but at lower pressures.
- In piping where the space is inadequate for conventional arrangement for providing flexibilities.

It is not advisable to use the expansion joint in all piping systems.

The major areas of applications where its use is not advisable are following piping systems.

- where hazardous chemicals are handled.
- where the service is high pressure.
- in which slurry or suspended solids are handled.

4. MATERIALS OF CONSTRUCTION

Based on the service for which the expansion joint is selected/used, the material of construction of the same is selected. Expansion joints are available in the following materials of construction.

- **Rubber**
- **PTFE**
- **Canvas**
- **Metal**

Except for the metallic expansions joints, all others are used to isolate the vibrating equipment from the ducting/piping. The selection is limited by pressure, temperature and the compatibility with the service fluid.

The rubber expansion joints are available with single convolution or multiple convolutions. Metallic split flanges are provided as retaining lugs to ensure pressure tight seal. These are provided with tie rods which restrict the lateral movements. To ensure that no damage is done to the expansion joint, pipes should be anchored at change in pipe direction, branching of pipe, change in pipe size and end of pipe run.

The PTFE expansion joints are mainly used in glass piping to ensure that no strain is transmitted to the pipeline. Such joints are provided at all equipment connections and change in direction on piping.

Expansion joints made from canvas are not suitable for liquid service. These are used in very low pressure systems and can be used in services where operating temperatures are moderate. These are mainly used in ducting to isolate the vibrating equipment.

Rubber, PTFE, Canvas expansion joints are designed and manufactured as per manufacturers' standard.

The metallic expansion joint is manufactured from austenitic stainless steel of required grade depending upon the service conditions. These are designed and manufactured as per E J M A (Expansion Joint Manufacturers Association) standard.

5. TYPES OF EXPANSION JOINT MOVEMENTS

It has been indicated that the expansion joint is used to absorb the movement in the pipeline. Let us consider the various possible movements to be absorbed by an expansion joint.

5.1 Axial Movement

The dimensional lengthening or shortening of a bellow parallel to its longitudinal axis is termed as the axial movement. (Refer Fig 5.1)

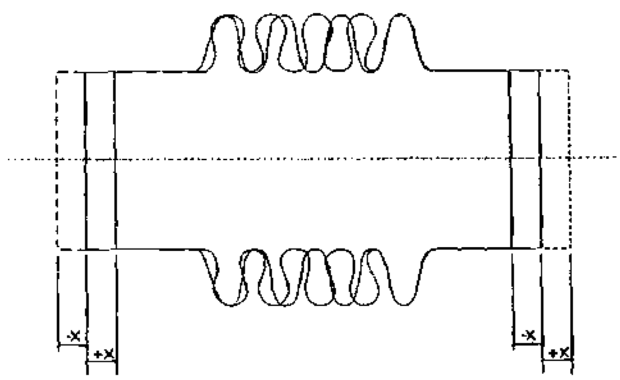


Fig. 5.1

5.2 Lateral Deflection

The displacement of one end of the bellow expansion joint relative to the other end, perpendicular to the longitudinal axis is termed as lateral deflection. (Refer Fig. 5.2)

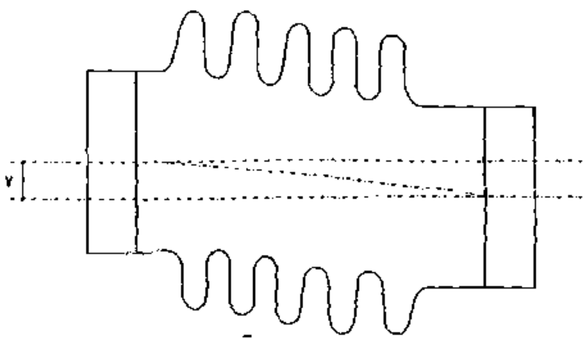


Fig. 5.2

The deflection could be multidirectional as well. (Refer Fig. 5.3)

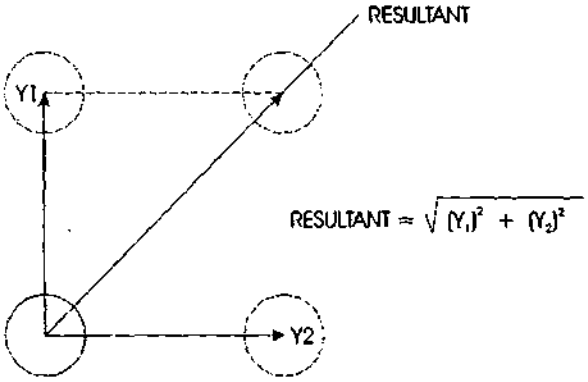


Fig. 5.3

5.3 Angular Rotation

The displacement of the longitudinal axis of the expansion joint from its initial straight line position to a circular arc is termed as angular rotation. (Refer Fig 5.4)

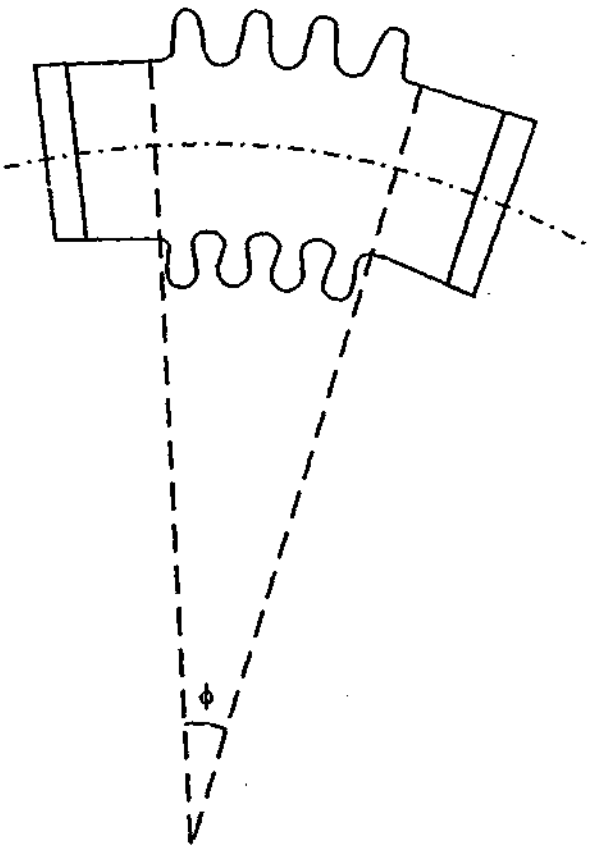


Fig. 5.4

5.4 Torsional Movement

The twisting of one end of the expansion joint with respect to the other end is called the torsional movement (Refer Fig. 5.5). Torsional movement imposes severe stresses in the expansion joint and special care has to be taken for the use of expansion joint for such application.

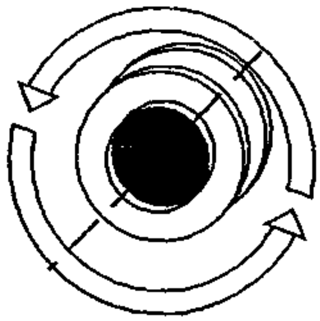


Fig. 5.5

6. COMPONENTS AND ACCESSORIES

Expansion joints are used in the industry for multiple applications. To perform such functions and to meet complex requirements of industrial applications, there are various additions to the basic components and accessories of an expansion bellow. (Refer Fig. 6.1 and 6.2)

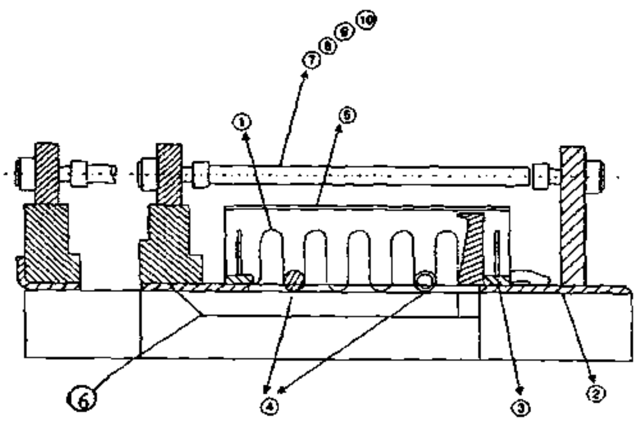


Fig. 6.1

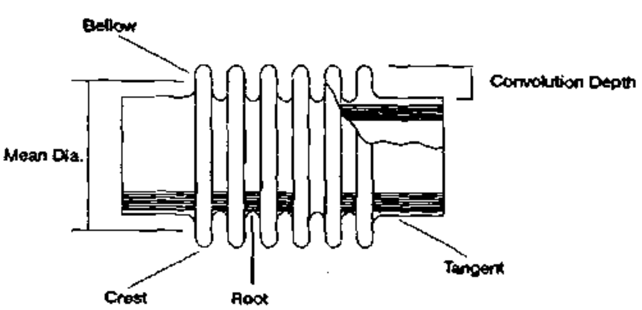


Fig. 6.2

6.1 Bellow

Bellow is the corrugated portion of the expansion joint responsible for absorbing movement. Bellow consists of a number of convolutions, which is directly proportional to the total movement to be absorbed by the bellows. The top most position of the convolution is called the crest and the bottom most portion is called the root. The depth of the convolution is the total height of the same and is the distance between the crest and the root. The average of the diameters of the crest and the root is termed as the mean diameter of the bellow. When the internal pressure demands a higher thickness of bellow, the flexibility gets reduced. To overcome this, bellows are made with multi ply thin wall sections. This will permit larger movements withstanding high pressure and will provide same service life.

6.2 Tangent

The straight portion at the two ends of the bellows on which the end connections are made is termed as tangent. The end connections could be provided with beveled joint to connect the expansion joint to the piping by welding. It could be provided with either welded or loose flanged connections depending on the piping specification.

6.3 Collar

This is a ring of suitable thickness by which the bellow is secured to the tangent. This prevents the bellow from bulging due to pressure.

6.4 Reinforcing Rings

When bellows are meant to be used for high pressure service, reinforcing rings made out of either tubing or solid bar will be fitted strongly to the roof of the convolutions. This is considered as a safety measure to ensure that the convolutions do not open out due to extra pressure that gets applied occasionally.

6.5 Lagging Shroud

This is an external cover provided over the bellows. In addition to providing protection to the bellows from mechanical damage, this also prevents the insulation material from entering the root of the convolutions which may prevent the bellow from functioning. This is also termed as cover or external shroud.

6.6 Internal Sleeves

This is a thin pipe section placed inside the bellows to prevent contact between the inner surface of the bellows and the fluid flowing through it. This device is provided to protect the bellows convolutions from damage due to resonant vibration when the fluid velocity is high. This will also prevent erosion when the service involves abrasive media. While installing a bellow with internal sleeve, care should be taken to mount the same in proper direction with respect to the direction of flow. The internal sleeve is sometimes referred to as liner as well.

6.7 Limit Rods

To restrict the movement of the bellow in axial, angular or lateral direction during the normal operation, solid rods are provided to space the bellow assembly. These are designed to prevent the bellows from over extension or over compression by dynamic loading generated due to the pressure loading. These are used in pressure balanced type of joints.

6.8 Tie Rods

These are solid rods or bars spacing the bellow assembly provided to restrict the axial movement and permitting the lateral deflection during the normal operation. These are suitably designed to absorb the pressure thrust due to internal pressure.

6.9 Shipping Devices

This device is provided as a protection against damages which can occur to the assembly during transportation. This will maintain the installation tight by keeping configuration of the convolution. Care should be taken to keep this device till the lines are hydro tested and removed prior to start up of the system.

6.10 Pantographic Linkages

This arrangement is done in hinged or gimbal type expansion joints. This is a scissor like device, which will allow rotational movement while restricting the axial and lateral movement of the bellow.

7. TYPES OF EXPANSION JOINTS

There are different types of expansion joints manufactured to take care of various requirements of the industry. The various types available are as follows:-

- **Axial - Single/ Double**
- **Universal**
- **Swing**
- **Hinged**
- **Gimbal**
- **Pressure Balanced**
- **Tied**

A brief description of these is given below.

7.1 Axial Expansion Joint

7.1a AXIAL- SINGLE EXPANSION JOINT

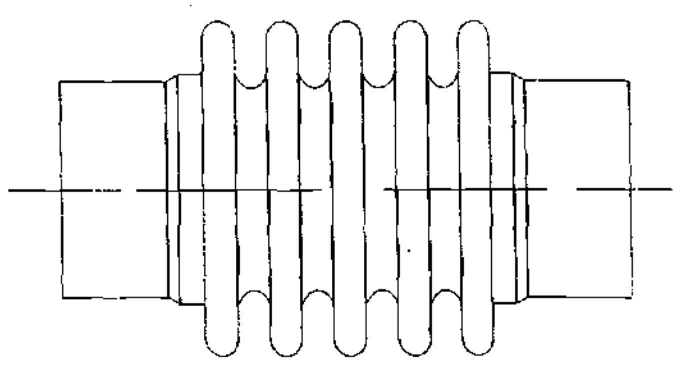


Fig. 7.1a

This is the simplest form of expansion joint of single bellows construction. It absorbs all of the movement of the pipe section into which it is installed. These bellows are capable of absorbing only small amounts of lateral or angular movement.

7.1b AXIAL- DOUBLE EXPANSION JOINT

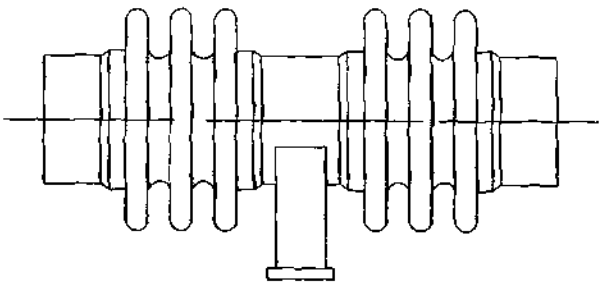


Fig. 7.1b

A double expansion joint consists of two bellows jointed by a common connector which is anchored to some rigid part of the installation by means of an anchor base. The anchor base may be attached to the common connector either at installation or at the time of manufacturing. Each bellow of a double expansion joint functions independently as a single unit. Double bellow expansion joints should not be confused with universal expansion joints.

7.2 Universal Expansion Joint

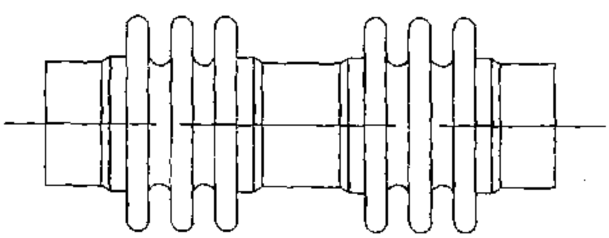


Fig. 7.2

A universal expansion joint is one containing two bellows joined by a common connector for the purpose of absorbing any combination of three (3) basic movements. A universal expansion joint is used in cases where it is necessary to accommodate greater amounts of lateral movement than can be absorbed by a single expansion joint.

7.3 Swing Expansion Joint

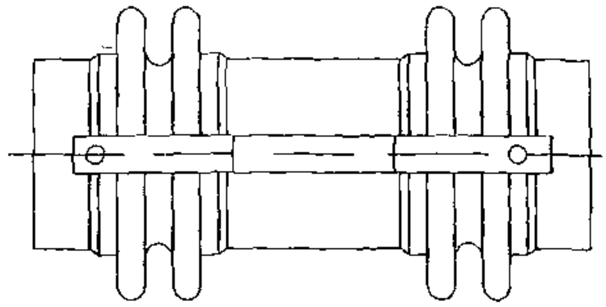


Fig. 7.3

A swing expansion joint is designed to absorb lateral deflection and/or angular rotation in one plane only by the use of swing bars, each of which is pinned at or near the ends of the unit.

7.4 Hinged Expansion Joint

A hinged expansion joint contains one bellow and is designed to permit angular rotation in one plane only by the use of a pair of pins running through plates attached to the expansion joint's ends. Hinged expansion joints should be used in sets of 2 or 3 to function properly.

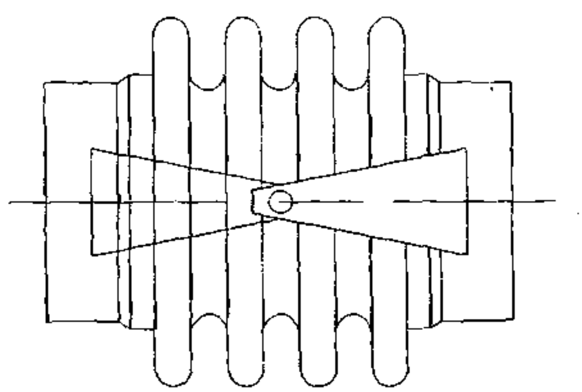


Fig. 7.4

7.5 Gimbal Expansion Joint

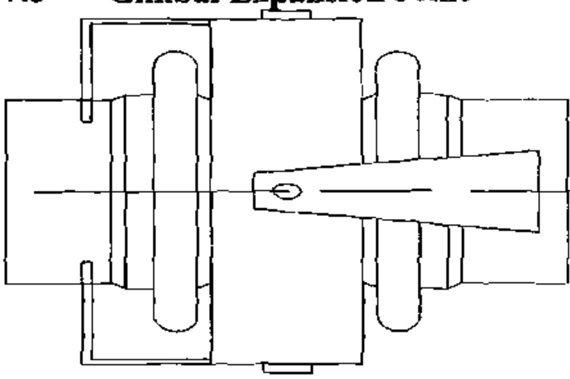


Fig 7.5

A gimbal expansion joint is designed to permit angular rotation in any plane by the use of two pairs of hinges affixed to a common floating gimbal ring.

7.6 Pressure Balanced Joint

A pressure balanced expansion joint is designed to absorb axial movement and/or lateral deflection while restraining the bellows pressure thrust force by means of the devices interconnecting the flow bellow with an opposed bellow also subjected to line pressure. This type of joint is installed where a change of direction occurs in a run of pipe, where it is not possible to provide suitable anchors.

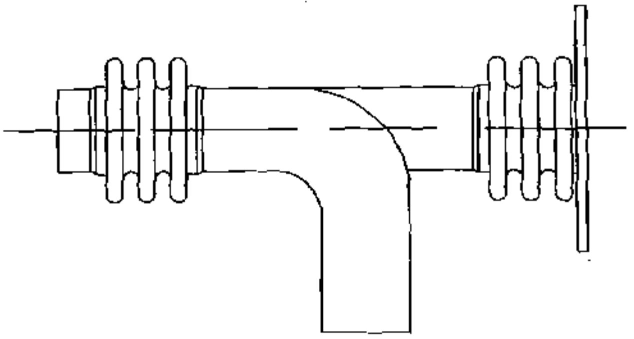


Fig. 7.6

7.7 Tied Expansion Joint

These are bellows provided with tie rods to restrict axial movement, while the bellow is subjected to high pressure services. Tie rods can be provided on single, universal or pressure balanced type of expansion joints.

Selection Chart

Sr. No.	Type of Expansion Joint	Axial Movement	Lateral Movement	Angular Rotation	Elimination of Pr. Thrust
1	Axial	Yes	No	No	No
2	Universal	Yes	Yes	Yes	No
3	Swing	No	Yes	Yes	Yes
4	Hinged	No	No	Yes	Yes
5	Gimbal	No	No	Yes	Yes
6	Pressure Balanced	Yes	Yes	No	Yes
7	Tied	No	Yes	No	Yes

The selection of a proper expansion joint involves a number of variables, including piping configuration, the operating conditions, derived cycle life, load limitations on the equipment etc. The major factor is the unique character available with the type of design, which makes it suitable for a particular application. The selection chart will facilitate a selection.

Before discussing the applications of the various types of expansion joints, it is required to derive formulas, which contain terms which are of common use and which are required for the proper selection and application of the various types of expansion joints.

8.0 GLOSSARY OF TERMS

8.1 Pipe Anchor

The purpose of anchor is to divide a pipeline into individual expanding/ contracting sections. The function of pipe anchor is to limit and control the movement with expansion joints located between the anchors absorbing the same.

8.2 Main Anchor

Main anchor is located at any of the following points in a pipe section.

- Between two bellow units installed on the same pipeline.
- Change in direction as at elbows when the advantage of elbow is not considered in flexibilities.
- At the entrance of a side stream into main pipeline i.e. in "T" section.
- At bend ends of pipe containing bellows.

Main anchor should be so designed that it is capable of withstanding forces and moments imposed by the pipe section between which bellows are located. When main anchor is installed at the pipe bend, the centrifugal thrust as a result of change in direction of flow should also be considered.

8.3 Intermediate Anchor

These are anchors provided in between the main anchor dividing the pipeline into individual expanding pipe sections.

8.4 Pipe Guides

Pipe guides are those, which permit axial movement while preventing angular or lateral movement. These are of significant importance for proper functioning of the expansion joint.

8.5 Directional Stop/Anchor

Directional stop/anchor is a device, which is designed to absorb loading in one direction while allowing the movement in another.

8.6 Spring Rate

This is a measure of bellows flexibility. It is the force required to extend or compress the bellow per unit length in the axial direction parallel to its longitudinal axis. It is expressed in kg/mm or lbs/in.

8.7 Spring Force

While absorbing the movements, the bellow imparts forces and moments to the piping system, which should be absorbed by proper provision of support and structures. This force is the product of the deflection absorbed and the spring rate of the bellow.

8.8 Pressure Thrust

This is the force due to internal pressure acting to open out the bellows. The magnitude of the pressure thrust is the product of the system pressure and the area at mean diameter of the bellow. In case of positive pressure, the convolutions are pushed apart causing the bellows to elongate while the case is reverse in the case of an external pressure.

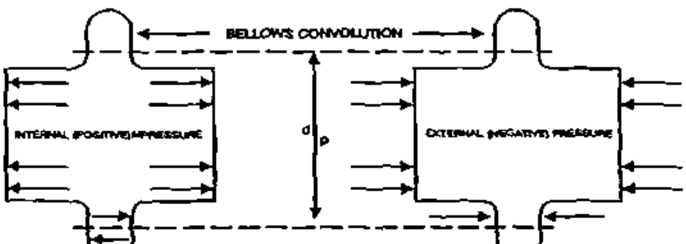


Fig 8.1

8.9 Cycle Life

This is defined as the number of movements an expansion joint is able to perform from the initial position to the operating position and then return to initial position before it fails.

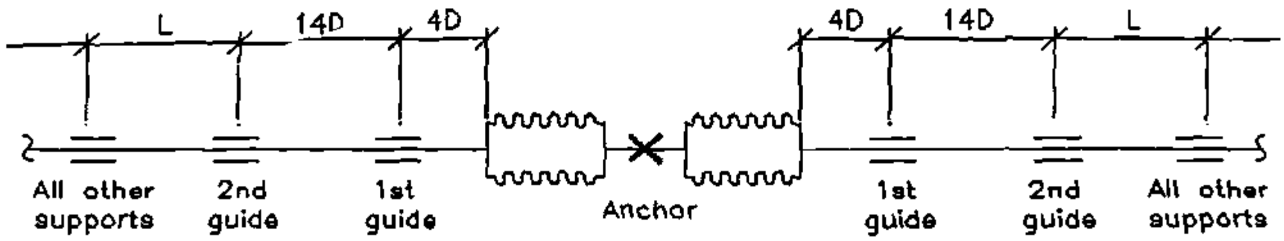


Fig. 9.1 Pipe Guide Location

9. APPLICATION

The knowledge of the application of the various types of expansion joints is important, as it is required for the selection. The location of anchors as well as guides are also very important for the proper functioning of the same.

The general guideline used in the location of guides is that the first pipe guide must be located within a distance of four pipe diameters from the end of the bellow and the second guide must be located within a distance of fourteen pipe diameter from the first guide. The subsequent support could be placed at the maximum span allowed as per the pipe size and the service for which it is meant for.

9.1 Axial Expansion Joints Tied/Untied

Fig. 9.2

The axial expansion joint could be single or double with an intermediate anchor. Because it offers the lowest expansion joint cost, the single expansion joint is usually considered first for any application.

Fig 9.2 shows a typical application of a single expansion joint absorbing combined axial movement and lateral deflection. The expansion joint is located at one end of the long piping leg with main anchors at each end and guides properly placed for both movement control and protection of the piping against buckling. In this case, however, the anchor at the left end of the line is a directional main anchor, which while absorbing the main anchor loading in the direction of the expansion joint axis, permits the thermal expansion of the short piping leg to act upon the expansion joint as lateral deflection. Because the main anchor loading exists only in the piping segment containing the expansion joint, the anchor at the end of the shorter piping leg is an intermediate anchor.

Fig 9.3

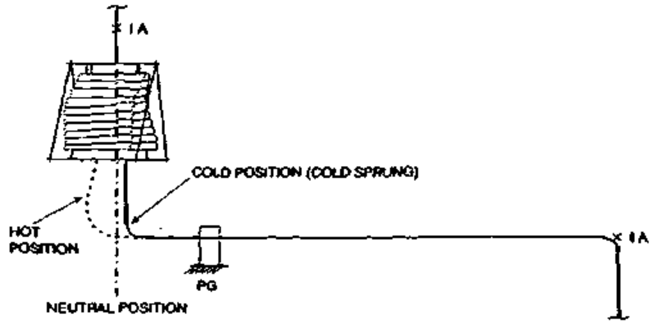


Fig. 9.5

Fig 9.3 shows an alternate arrangement in which the expansion joint is installed in the short piping leg and the principal expansion is absorbed as lateral deflection.

Note that in this case, the longer piping leg is free of compressive pressure loading and requires only an intermediate anchor and directional guiding. The functions of the directional anchor and the pipe guide may be combined in a single device.

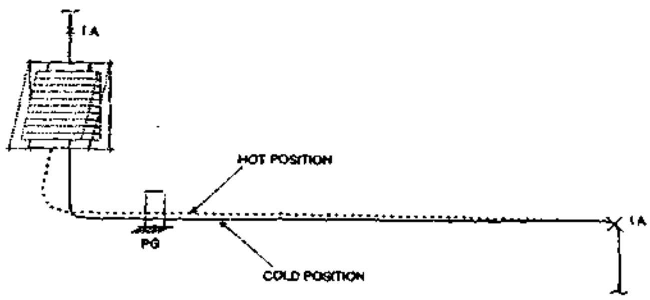


Fig. 9.4

Fig. 9.4 and 9.5 represent modifications over Fig. 9.3 in which the main anchors at either end of the expansion joint are replaced by tie rods. Where the piping configuration Permits, the use of tie rods frequently simplifies and reduces the cost of the installation. Because of these tie rods, the expansion joint is not capable of absorbing any axial movement other than its own thermal expansion. The thermal expansion of the piping in the shorter leg is, as a result, imposed as deflection on the longer piping leg. In some cases, where the longer piping leg is not sufficiently flexible and where the dimension of the shorter leg is suitable, the rods may be installed spanning the entire short leg so that no deflection is imposed on the longer run from this source.

Where appreciable amounts of lateral deflections are imposed upon the expansion joint, some shortening of the expansion joint results from the displacement of the tie rods as shown in Fig. 9.4

Care should be taken to insure that sufficient piping flexibility exists to absorb this deflection and that adequate clearances are provided in the guide to permit deflection of the piping. The amount of this deflection can be minimized by cold springing the expansion joint in the lateral direction as shown in Fig.9.5

The principal restriction upon the use of single expansion joint for lateral deflection or combined axial movement and lateral deflection is the limited amount of lateral deflection, which such an expansion joint can absorb. The allowable lateral deflection is directly proportional to the ratio of corrugated length to diameter,

which, in turn, is restricted by considerations of stability and manufacturing limitations. Thus, while eminently suitable for applications such as Fig 9.2 where the principle movement is axial, the relatively small available lateral movement severely limits the type of application illustrated in Fig 9.3, 9.4 and 9.5. Where operating pressures and temperatures are high, or where availability of suitable structures precludes the use of main anchors and multiple guides, the application shown in Fig 9.2 may not be feasible and another type of expansion joint may result in a far more economical installation.

9.2 Universal Expansion Joints

The universal expansion joint is particularly well adapted to the absorption of lateral deflection. In addition, this design may be used to absorb axial movement, angular rotation or any combination of the three. The most common application of the universal expansion joint is its use as a tied expansion joint in a 90 degree piping offset., with the tie rods adjusted to prevent external axial movement. Two such applications are shown in Fig. 9.6 and 9.7

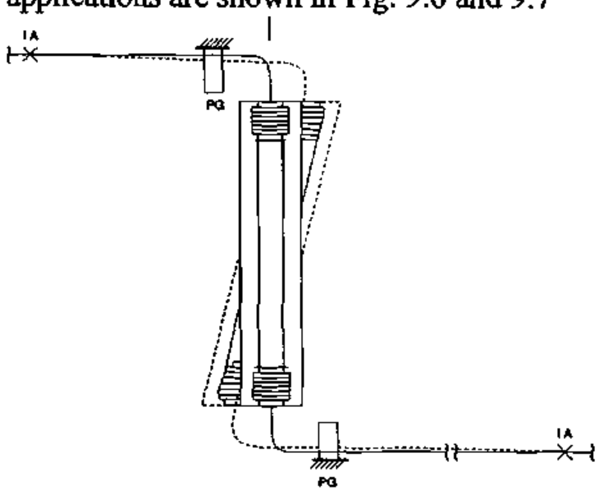


Fig. 9.6

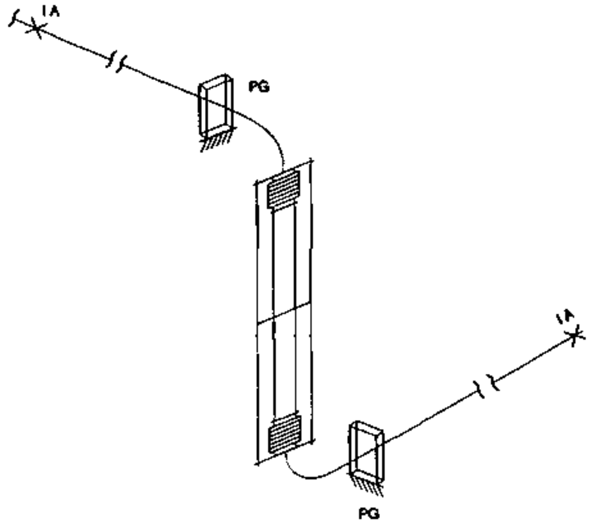


Fig. 9.7

Fig 9.6 shows a tied universal expansion joint used to absorb lateral deflection in a single plane "Z" bend. Where dimensionally feasible, the expansion joint should be designed to fill the entire offset leg so that its expansion is absorbed within the tie rods as axial movement. The thermal movement of the horizontal lines is absorbed as lateral deflection by the expansion joint.

Both anchors are intermediate anchors since the pressure loading is absorbed by the tie rods. Only directional guiding is required since the compressive load on the pipe consists only of the force necessary to deflect the expansion joint. Any thermal expansion of the offset leg external to the tie rods, such as that of the elbows at either end, must be absorbed by bending of the horizontal pipe legs.

Provision should be made in the design of the guides to allow for both this deflection and the reduced length of the expansion joint in its deflected position. In addition, particularly in the case of long universal expansion joints under high pressure, additional allowances may be necessary to compensate for stretching of the tie rods under load. The expansion joint manufacturer should be consulted for

recommended minimum guide clearances.

Fig 9.7 shows a typical application of a tied universal expansion joint in a three-plane "Z" bend. Since the universal expansion joint can absorb lateral deflection in any direction, the two horizontal piping legs may lie at any angle in the horizontal plane.

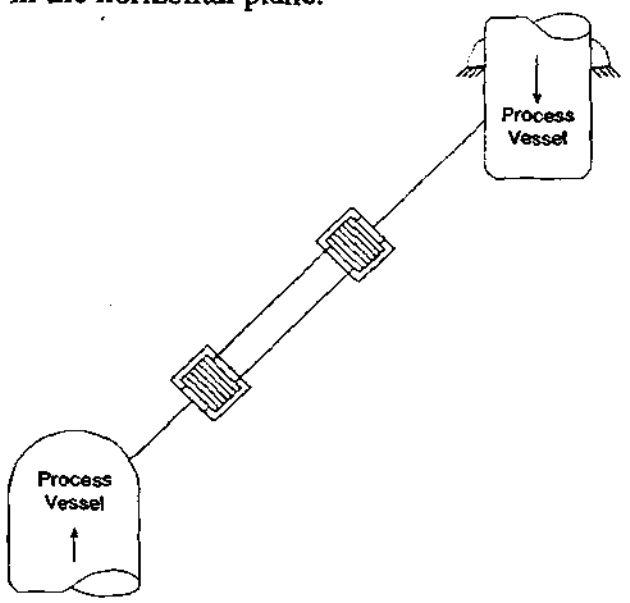


Fig 9.8

In cases where a universal expansion joint must absorb axial movement other than its own thermal growth, it cannot function as a tied expansion joint and must be used in combination with main anchors to absorb pressure loading. One such case is shown in Fig 9.8

The relative expansion between the two vessels results in both axial movement and lateral deflection on the expansion joint. Both vessels must be designed to absorb main anchor loading. Limit rods may be used to distribute the movement between the bellows and to control their movements.

As a direct result of increasingly high operating pressures and temperatures, and lighter building construction methods, universal expansion joints are finding increasing use in steam and hot water distribution systems where, due to their ability to absorb large amounts of movement with minimum guiding and anchoring, they offer impressive savings in overall cost.

Numerous variations are possible in the design of universal expansion joints. In a horizontal installation, for example, where it is desirable to support the center pipe section of the expansion joint independently of the bellows, tie rods or external structural members may be used. In a single plane system, the tie rods may be placed by two bars with pinned connections at either end of the expansion joint. This construction is so commonly used that it has been given the standard nomenclature of "swing expansion joint". In some cases, two sets of short control rods, one spanning each of the two bellows in the universal expansion joint, are used instead of the overall tie rods shown in most of the illustrations. This arrangement is frequently used where the expansion joint must absorb axial movement and where the control rods are used primarily for control and stability rather than for absorption of pressure loading.

Where the universal expansion joint is very long in relation to its diameter, where a large number of corrugations are used at each end of the expansion joint or where the expansion joint is subject to external forces such as wind loading, vibrations, etc. it may be desirable to incorporate control devices in the expansion joint to prevent excessive displacement of the bellows and the relatively free pipe section between them.

Fig. 9.9 and 9.10 show two forms of controls which may be used for this purpose. In Fig 9.9, short rods are used spanning each of the bellows in the expansion joint. Stops are provided on the rods so that, once the expansion joint has reached its rated lateral deflection, the stops will be engaged by members rigidly fastened to the pipe

portions of the expansions joint and no further displacement will be possible.

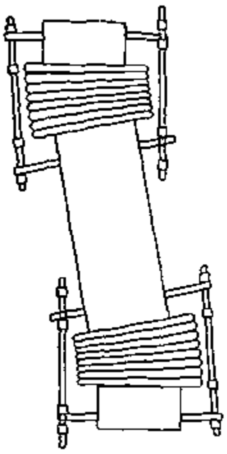


Fig. 9.9

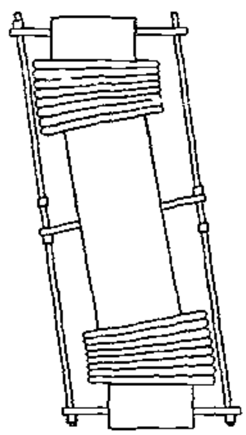


Fig.9.10

Fig 9.10 shows a similar device adapted to an expansion joint with overall tie rods. In this case, the rods' tops are engaged by a plate or lug attached to the center pipe portion and movement of this part beyond its design deflection is prevented. In order to obtain maximum control from these devices, the stops are usually oriented to lie in the plane of resultant movement of the expansion joint, affording maximum leverage as well as greater sensitivity to small movement. Devices of this nature are usually stipulated by the manufacturer when the design characteristics of the expansion joint warrant.

Despite the versatility of the universal expansion joint, its use is sometimes precluded by the configurations of the piping, the operating conditions or even by manufacturing and transportation limitations. Where, for example, the length of the offset leg in a "Z" bend is extremely long, it may be undesirable or impossible to fabricate, ship to the job site and install a universal expansion joint which would span the full length of the offsets. Further, where the expansion joint is very long in relation to its diameter, the flexibility of overall rods may reduce the effectiveness of the control so that the center pipe section becomes unstable. Where such limits are encountered, other types of expansion joints may offer a more desirable solution.

9.3 Pressure Balanced Expansion Joints

The pressure balanced expansion joint is used most frequently in applications similar to those shown for the single expansion joint, but where pressure loading upon piping or equipment is considered excessive or objectionable. The major advantage of the pressure balanced design is its ability to absorb externally imposed axial movement without imposing pressure loading on the system. It should be noted, however, that the force required to move the expansion joint is not balanced. In fact, it is increased over that of a single expansion joint. Since both the flow bellows and the balancing bellows must be compressed or elongated, the combined axial force acts upon the piping or equipment. Since the forces to move the bellows are generally of a low order of magnitude, these are usually not objectionable, except in cases involving extremely light equipment with close clearance moving parts which might be affected by small forces.

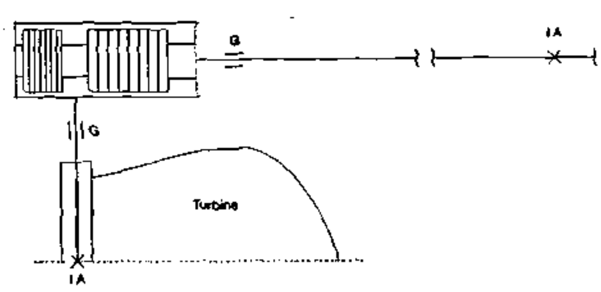


Fig 9.11

Fig 9.11 shows a typical application of a pressure-balanced expansion joint for combined axial movement and lateral deflection. Both the anchor at the end of the

piping run and that on the turbine are intermediate anchors and only directional guiding is required. By proper design, the guide directly above the turbine can be made to absorb the axial movement forces of the expansion joint without imposing these on the turbine. The only force imposed on the turbine is that which is required to deflect the expansion joint laterally.

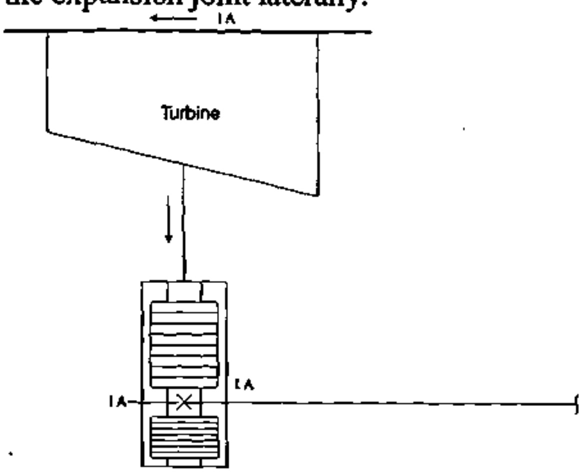


Fig 9.12

Fig 9.12 shows another turbine application, but, in this case, the anchor point of the turbine is located some distance from the expansion joint and the expansion of the turbine between its anchor and the expansion joint is absorbed as lateral deflection. An intermediate anchor is used at the center fitting of the expansion joint. Since the expansion joint is located close to the turbine, guiding between the turbine and expansion joint is not required.

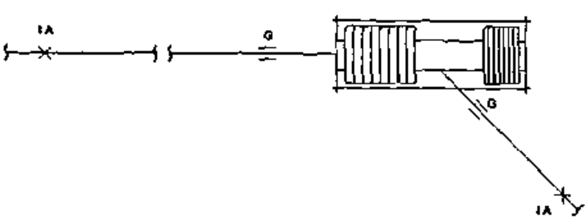


Fig 9.13

Figure 9.13 shows that a pressure balanced expansion joint can be used at changes in direction other that 90 degrees. In this case, the growth of the longer piping run is absorbed as axial movement on the expansion joint, while the thermal expansion of the offset piping run introduces both axial and lateral components of deflection on the expansion joint. Again, only intermediate anchors are required at the ends of the lines and directional guiding is used. The guide on the offset run may be used to absorb the axial movement forces of the expansion joint, if the piping is not sufficiently stiff to transmit this directly to the intermediate anchor.

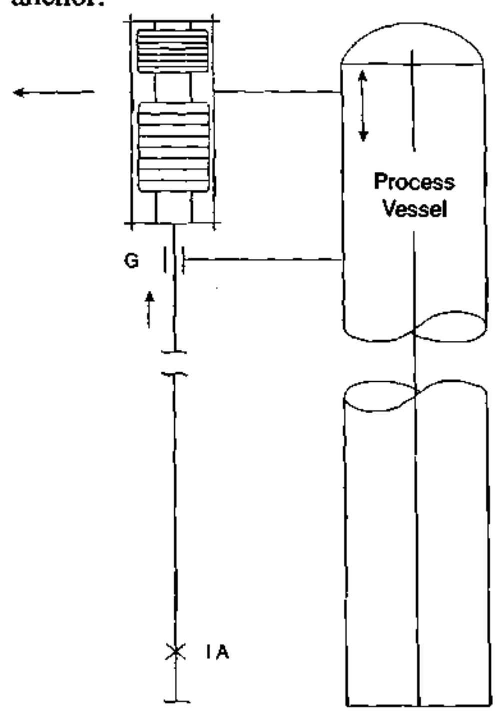


Fig 9.14

Fig 9.14 shows a common application for which a pressure balanced expansion joint is well suited. Under various process conditions, the vessel and the vertical pipe may expand at different rates. By installing a pressure balanced expansion joint, as shown, the differential vertical movement is absorbed as axial movement in the expansion joint and the thermal expansion from the center line of the process vessel to the piping is absorbed as lateral deflection.

The piping may then be secured by an intermediate anchor at the bottom and furnished with a directional guide adjacent to the expansion joint. as shown. In many cases no external structure is available at the upper elevation of the process vessel and the guide must be connected to the vessel itself. Using this arrangement may especially help where the vessel is tall and is subject to wind loading deflection, or similar effects where the guide is attached to a rigid external structure. The expansion joint must be designed to absorb wind loading deflection, etc., as lateral deflection are involved, Pressure balanced universal expansion joints are used in the flow end of the expansion joint and a single bellow in the balancing end. Normally, as shown in Fig. 9.15, the balancing bellows will be subjected to axial movement only if the tie rods are properly designed to rotate or pivot at their attachment points.

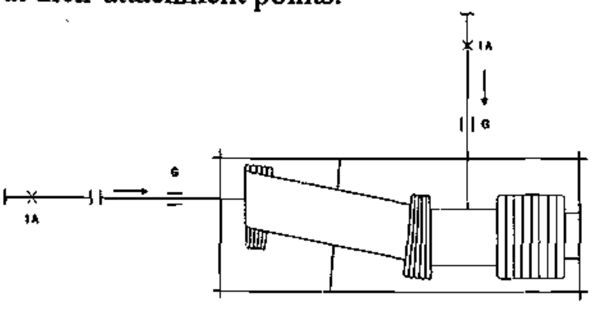


Fig 9.15

In order for a pressure balanced expansion joint to function properly, the pressure thrust restrained by the tie rods must exceed the axial movement forces of the expansion joint. In a large diameter, low pressure application, it may be impossible to utilize the pressure balanced expansion joint to eliminate the pressure loading or at best, the effect may be uncertain. In such cases, some other expansion joint design must be considered.

Pressure balanced expansion joints are not recommended for use in services where the pressure equalizing connections between the flow bellows and the balancing bellows may become plugged or blocked by the flowing medium or contaminants. Where flow considerations permit, this problem may be overcome by the use of a tee as a center fitting of the expansion joint rather that an elbow. In some cases, the pressure for the balancing end of the expansion joint has been introduced from a separate pressure source. A control failure or even a slow control response might result in partial or full pressure loading being imposed upon the piping or equipment, thus defeating the initial reason for using the pressure balanced expansion joint.

From the view point of cost, it must be considered that the pressure balanced expansion joint requires the use of an extra bellow which does not add to its ability to absorb movement. In addition, the expansion joint is usually furnished with a center fitting, either elbow or tee, which would otherwise constitute a portion of the piping cost. Further, the necessary structure i.e blind flanges, tie rods and attachment structures add appreciably to the cost of the expansion joint. The use of a pressure balanced expansion joint can be justified economically only where the problems created by the pressure loading represent an even greater cost.

The pressure balanced expansion joint is finding increasing use for the sole function of relieving loads upon equipment such as pumps, compressors and turbines. In many cases, the cost of the pressure balanced expansion joint will be negligible when compared to the cost of additional equipment, piping and building space which would be necessary for safe functioning of the equipment without the expansion joint.

9.4 Hinged Expansion Joints

Hinged expansion joints are usually used in sets of two or three, to absorb lateral

deflection in one or more directions in a single plane piping system. Each individual expansion joint in such a system is restricted to pure angular rotation by its hinges. However, each pair of hinged expansion joints, separated by a segment of piping will act in unison to absorb lateral deflection in much the same manner as a swing or universal expansion joint in a single plane application. For a given angular rotation of the individual expansion joint, the amount of lateral deflection which a pair of hinged expansion joints can absorb is directly proportional to the distance between hinge pins. Therefore, in order to utilize the expansion joints most efficiently, this distance should be made as large as possible. Expansion joint hinges are normally designed to absorb the full pressure thrust of the expansion joint and, in addition, may be designed to support the weight of piping and equipment, wind loads or similar externally applied forces. Where such external forces are anticipated, their direction and magnitude must be indicated to the expansion joint manufacturer so that the hinges can be adequately designed to withstand these forces.

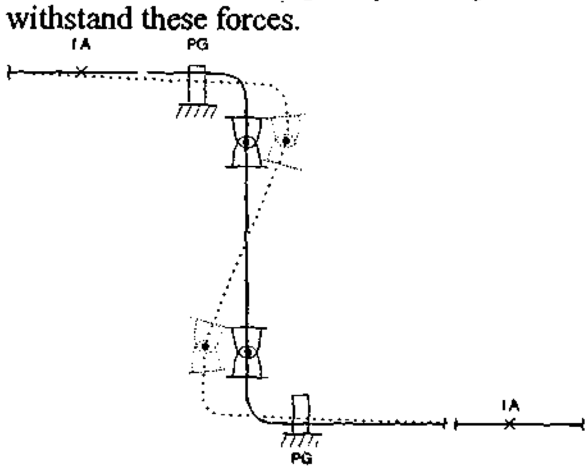


Fig 9.16

Fig 9.16 illustrates the use of a two hinge system to absorb the major thermal expansion in a single-plane "Z" bend. Since the pressure thrust is absorbed by the hinges on the expansion joints, only intermediate anchors are required at each end of the piping system. The thermal expansion if the offset section containing the expansion joints must be absorbed by bending of the piping legs perpendicular to that segment, since the expansion joints are restricted to pure angular rotation by their hinges and cannot extend or compress. The amount of bending deflection imposed on each of the two long piping legs may be controlled by proper design of guides and supports.

Where one long leg is sufficiently flexible to absorb the full thermal growth of the offset leg, the other long leg may be controlled to permit longitudinal movement only. The planar guides shown at the ends of the long piping runs near the elbow are intended to maintain the planarity of the piping system only and must of course allow for the bending deflections of the long piping legs. In calculating guide clearances, consideration should be given to the fact that the thermal expansion of the offset piping leg containing the expansion joints will be partially offset by the reduction in length resulting from the displacement of the center pipe section. The latter effect may be neglected only where the distance between hinge pins is very large and the lateral displacement small. This effect can be minimized by cold springing the expansion joints 50% of the full rated deflection.

Because of the ability of the hinges to transmit loads, support of a hinged piping system can frequently be simplified. Assuming that Fig. 9.16 is an elevation view, for example and that the upper piping leg is sufficiently flexible to absorb the total expansion of the vertical leg, it would be possible to use sliding supports on the lower horizontal run to support its weight and restrict it to longitudinal movement only. By utilizing the rigidity of the hinges, a substantial portion of the weight of the

upper horizontal leg may also be carried on these lower supports. It should be noted, however, that the sliding support nearest to the vertical leg must be designed to resist the force required to deflect the piping. Spring supports must be used throughout the length of the upper horizontal leg where bending occurs. Beyond that point, sliding supports may be used.

In locating hinged expansion joints for more efficient use, it should be noted that the hinges need not be collinear in order to function properly.

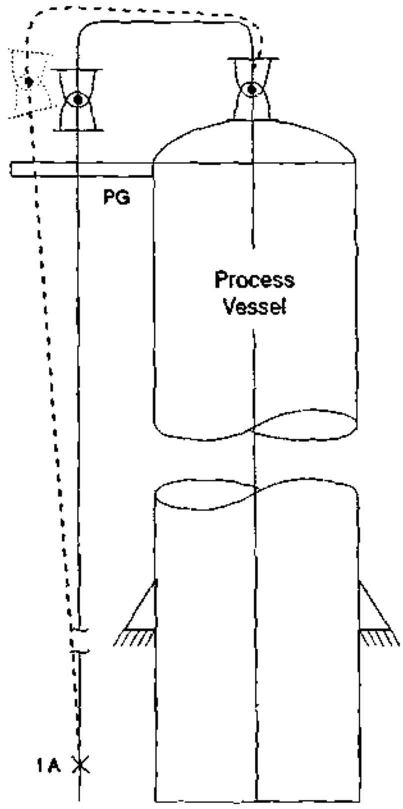


Fig 9.17

Fig 9.17 illustrates a two-hinge expansion joint system similar to the pressure balance expansion joint application of Fig 9.14. In this case, the expansion joints will absorb only the differential vertical growth between the vessel and the pipe riser. Any horizontal movement due to piping expansion, vibration, wind loads etc., will be absorbed by the bending of the vertical pipe leg. A planar guide may be installed near the top of the vessel to protect the hinged expansion joints from wind loads at right angles to the plane of the piping.

The anchor shown at the bottom of the riser is an intermediate anchor only. The pressure load is absorbed by the expansion joint hinges. However, this anchor must be capable of withstanding the forces created by bending of the riser. Depending upon the dimensions and weight of the piping system, complete support may be obtained from the process vessel and from the intermediate anchor. If additional supports are required , spring type supports should be used. If desired, the vertical piping may be cold sprung to reduce bending stresses, utilizing the hinges to withstand the cold spring force.

Where the piping in a single plane system is not sufficiently flexible to absorb the bending deflections involved in a two hinge system, or where the loads resulting from such bending exceed the allowable limits for connected equipment, a system of three hinged expansion joints may be used.

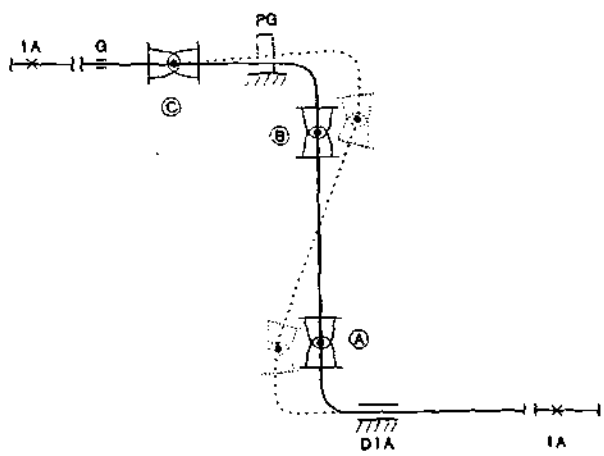


Fig 9.18

Fig 9.18 illustrates a system of three hinged expansion joints in a single plane "Z" bend. The thermal expansion of the offset piping section is absorbed by the action of

expansion joints B and C. It is therefore evident that expansion joint B must be capable of absorbing the total of the rotations of expansion joints A and C. Hence, it is frequently necessary that the r expansion joint at the center contain a greater number of corrugations than those at either end.

As in the previous cases, the anchors at the ends of the piping system are intermediate anchors only. In this case, all deflection is absorbed by the expansion joint and no pipe bending loads will be imposed upon these anchors. Where the distance between the anchor at the left and the first hinged expansion joint C is large, a pipe guide should be installed adjacent to the expansion joint, as shown in Fig.9.18. This pipe guide will minimize bending of the pipe section between expansion joint C and the left hand anchor, which might otherwise result from the moment required to rotate the expansion joint. One or more additional guides may be used to maintain the planarity of the piping system and relieve the hinges of bending forces, which may be created by external loads. Support of the piping system may be effected in various ways, utilizing available supporting structures with greatest efficiency. Here again, however, it is essential that spring supports be used to permit free movement of the piping between the expansion joints.

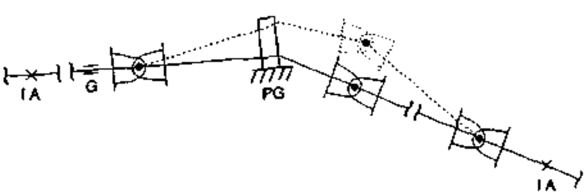


Fig.9.19

Fig 9.19 illustrates the principle that systems of hinged expansion joints may be used in other than 90° bends. In such applications, a three hinge system is usually most suitable, since the components of movement may be quite large and excessive bending stresses would result from the use of a two hinge system. Except for this point, the system is similar in every respect to the previous ones containing 90° bends. Only intermediate anchors and planar guides are required.

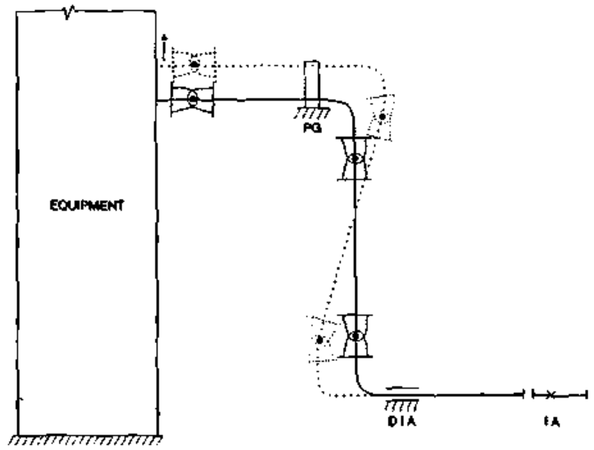


Fig. 9.20

A hinged expansion joint system may be used effectively in applications involving movement other that the pure thermal growth of piping. Fig. 9.20 illustrates an application combining the thermal expansion of piping system with the single plane movements of a piece of connected equipment. So long as all movements are restricted to a single plane, the behavior of the expansion joint system is quite similar to that of the system shown in Fig. 9.18. In this case, an intermediate anchor is required at one end of the piping. The equipment serves as an intermediate anchor at the opposite end. The displacements of the equipment are totaled with those of the piping in order to evaluate the movements of the expansion joints. Planar guide clearances in the plane of the piping must be adequate to allow for the equipment movement as well as for the piping rotations.

Among the major advantages of hinged expansion joints are their compact size,

which facilitates installation, and the great rigidity and strength which can be incorporated into the hinge structures. By the use of these individual units, it is frequently possible to compensate for the thermal expansion of irregular and complex piping configurations which might preclude the use of other types of expansion joints. Because of the ability of the hinge structure to transmit loads, piping system containing hinged expansion joints impose minimum force on the pipe anchors. Furthermore, such systems may be supported at virtually any point which does not interfere with the free movement.

9.5 Gimbal Expansion Joints

Just as hinged expansion joints may offer great advantages in single plane applications, gimbal expansion joints are designed to offer similar advantages in multiplane systems. The ability of the gimbal expansion joint to absorb angular rotation in any plane is most frequently applied by utilizing two such units to absorb lateral deflection. An application of this type is shown in figure 9.22. Since the pressure loading is absorbed by the gimbal structure, intermediate anchors only are required. Planar guides are provided to restrict the movement of each piping leg. As in the case of hinged expansion joints, the location of pipe supports is simplified by the load carrying ability of the gimbal structure. Since, in a two gimbal system, the growth of the vertical pipe leg will be absorbed by bending of the long legs, spring supports may be required on either or both of these. Guides must be designed to allow for the thermal expansion of the leg containing the expansion joints and for the shortening of this leg due to deflection.

Where it is impossible or undesirable for the piping to absorb the growth of the offset leg, a system consisting of two gimbal and one hinged expansion joints may be used as shown in Fig. 9.21.

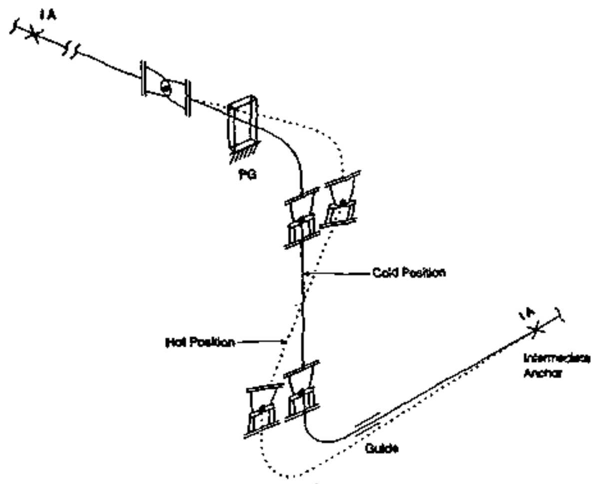


Fig. 9.21

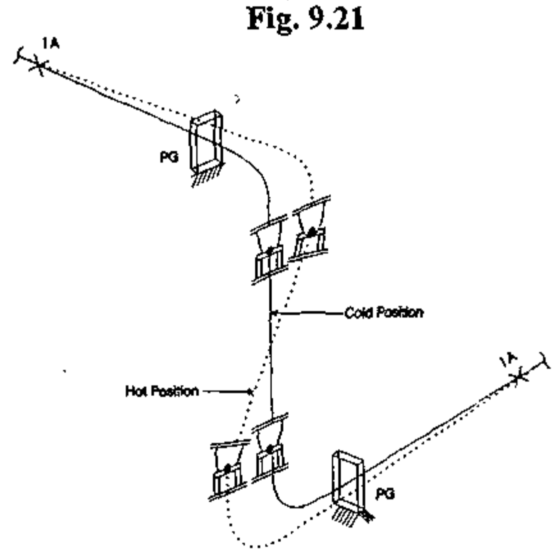


Fig. 9.22

The gimbal expansion joints function in unison to absorb the combined movements of the upper and lower legs, while the hinged expansion joints and the upper gimbal expansion joint act in combination to absorb deflection of the offset leg. Since the expansion of the offset leg takes place in one plane only, the use of the simpler hinged expansion joint is justified. The advantages of using gimbal expansion joint system are simpler to those previously mentioned for systems containing hinged expansion joints. Greater flexibility of usage is, however possible since gimbal expansion joints are not restricted to single plane systems.

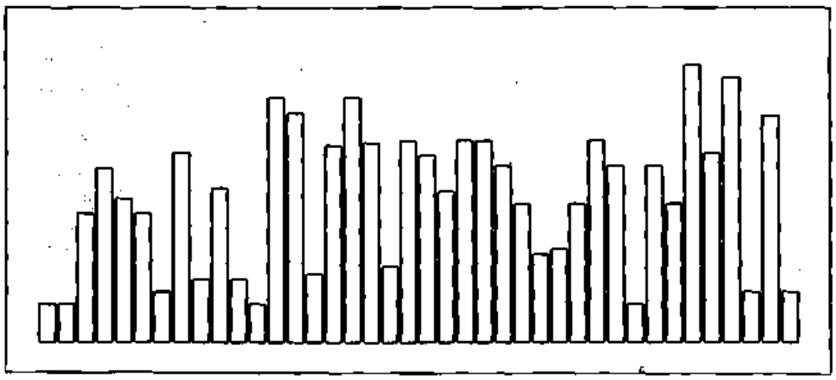
Certificate Course
on

PIPING ENGINEERING

June 12 - 25, 2006

DESIGN OF JACKETED PIPING

T. N. Gopinath
Consultant



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DESIGN OF JACKETED PIPING

T.N.GOPINATH

1.0 GENERAL

In a Chemical Process Industry transportation of material, especially in fluid form, poses a variety of problems. The problems are more when the fluid is viscous and has to be maintained at higher temperatures than the ambient throughout transport. Variations in ambient temperature from winter to summer also affect the flow characteristics of the liquid. The fluid inside the pipe can also undergo phase changes and the viscosity can change to adversely affect the fluid flow pattern. If it is only the atmospheric changes that create problems, then the temperature variation in the pipe is kept within the acceptable range by heat tracing. If the fluid has to be kept at a certain temperature all throughout the process of transportation, then the pipelines need jacketing. When a pipe of higher diameter is put over the service pipe, and when heating/cooling medium passes, as required, through the annular space created between the two pipes, then it is termed as a jacketed pipe. The inner pipe is called the core pipe and the outer pipe is called the jacket.

The combination of core and jacket pipes shall be selected based on :

i) The properties of the heating/cooling medium.
ii) The flow required to maintain the temperature.
iii) The criticality of the service.
iv) The differential expansion of the core and jacket when the material of construction of core and jacket are different.

The jacketed pipe poses problems, in design, fabrication and erection, different from that of the non-jacketed piping. This article is intended to highlight the problems of mechanical design of jacketed piping.

2.0 MECHANICAL DESIGN OF JACKETED PIPING

2.1 Thickness of core pipe:

There are two numbers of pipes involved in jacketed piping with different design considerations. The core pipe is subjected to internal pressure when there is fluid flow through the same. The pipe is subjected to external pressure when there is fluid flow through the jacket. It could be that these pressures get balanced and the pipe gets relaxed. But for mechanical design, the worst condition is to be considered. For any cylindrical surface subjected to external pressure, design is more complicated than for those subjected to internal pressure.

Code ASME B31.3 under clause 304.1.3 specifies that to determine the wall thickness for straight pipe under external pressure, the procedure outlined in the BPV Code Section VIII Division 1,UG-28 through UG-30 shall be followed.

Tracing material-
steam, hot water/oil
(2 to 3 bar-140 to 150°C)
LSHS - low sulpher heavy stage

When standard piping details are prepared the design length L cannot be exactly predicted. Hence the length to diameter (L/D) ratio of 50 is considered as a standard practice.
Since the pipes are manufactured with standard thicknesses, the selection has to be done from the available thickness ranges. The thickness considered for calculation should be after allowing for the mill tolerance and the corrosion allowance.

2.2 Size Combination of Core and Jacket Pipe:

i) *Straight Pipe:*

The size combination of the Core and jacket pipes are determined by the annular space necessary to obtain the required flow to maintain the heat transfer. In the absence of any specific process data, the following combinations are most widely used:

Size of Core Pipe NB (mm)	15	20	25	40	50	65	80	100	150
Size of Jacket Pipe NB (mm)	40	40	50	65	80	100	100	150	200

ii) *Elbows:*
The bending radius of elbows/bends are so selected that the core pipe and the jacket pipe maintain the same centrelines even at the change of direction. The ideal combinations of the core and jacket bends shall be as follows: -

Core pipe NB (mm)	Bends Radius (mm)	Jacket Pipe NB (mm)	Bend Radius (mm)	Remarks
15	60 *4D*	40	57 *1.5D*	NOTE 1
20	60 *3D*	40	57 *1.5D*	NOTE 1
25	75 *3D*	50	76 *1.5D*	NOTE 1
40	57 *1.5D*	65	62 *1.0D*	NOTE 2
50	76 *1.5D*	80	76 *1D*	NOTE 2
65	95 *1.5D*	100	102 *1D*	NOTE 2
80	114 *1.5D*	100	102 *1D*	NOTE 2
100	152 *1.5D*	150	152 *1D*	NOTE 2
150	229 *1.5D*	200	203 *1D*	NOTE 2

NOTE: i) Use 1.5D(LR) std. elbow for jacket.
ii)Use 1.5D(LR) std. elbow for core and 1D(SR) std elbow for jacket. (Refer Fig. 2.1)

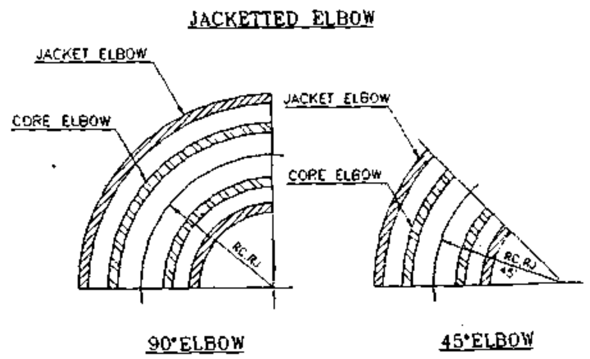


Fig. 2.1

2.3 Thickness of Jacket Pipe:

Jacket pipes have to be designed for internal pressure, which the jacket fluid exerts. The formula given in clause 319.4.1 of the code shall be followed with the standard procedure.

2.4 Types of Jacketing:

Depending upon the criticality of the requirement, the jacketing of the piping system can be done:

(i) Only on straight pipe keeping all bends and flange welds exposed.

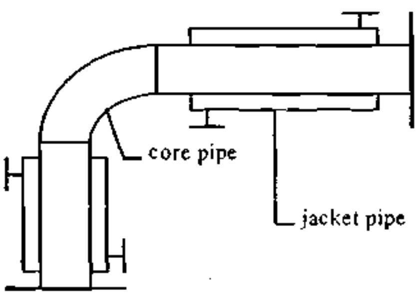


Fig. 2.2 a

(ii) On straight pipes and elbows but keeping the flange size same as that of the core pipe

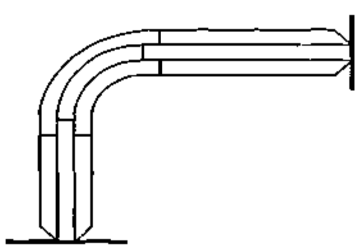
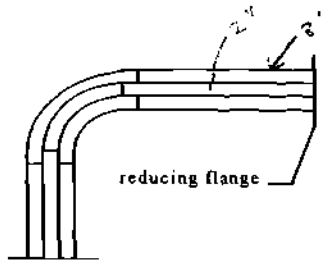


Fig. 2.2 b

(iii) On straight pipes and elbows with flange size that of the jacket pipe. (Reducing flanges)



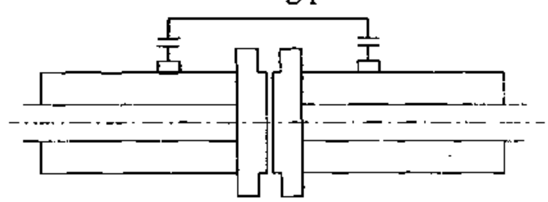
(then use Reducing Flange of 3" x 2")

Fig. 2.2 c

This type is used where jacketing requirements are critical. The slip on type flanges are modified to get a seating of the jacket pipe to achieve a proper welding joint. (Refer Fig. 2.7)

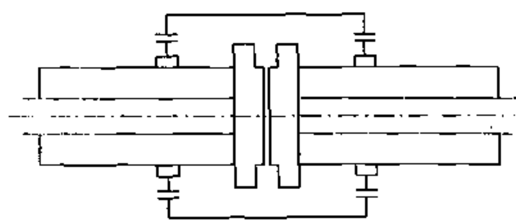
2.5 Jumper Pipes:

To maintain the continuation of fluid flow in the jacket, jumper pipes are provided. The location of jumper pipes on the horizontal jacketed pipe is decided based on the type of fluid in the jacket pipe. There can be a single jumper or two jumpers and these can be placed in one of the following patterns:



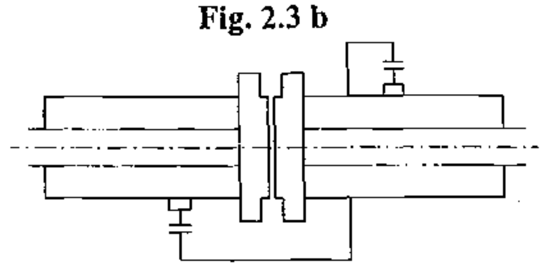
Pattern I

Fig. 2.3 a



Pattern II

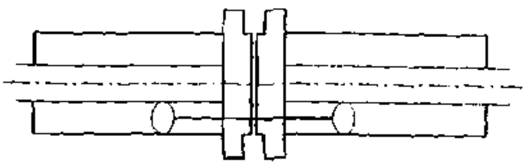
Fig. 2.3 b



Pattern III

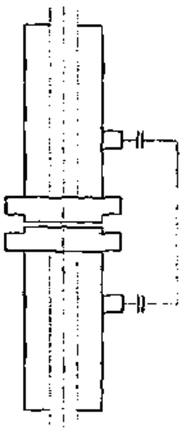
Fig. 2.3 c

Jumper size — for steam line → 1/2"
oil line → 3/4"



Pattern IV

Fig. 2.3 d



Pattern V

Fig. 2.3 e

When the heating fluid is in vapor form and no condensation is expected, arrangement as per Pattern I can be used. The arrangement in Pattern II makes the vapor phase as well as the liquid phase continuous and is ideal when steam is used as a heating medium in the jacket. When the jacketing fluid is a liquid under sufficient pressure, arrangement as per Pattern III or Pattern IV can be used. The arrangement for connection as per Pattern IV is difficult to fabricate, as the hole on the jacket pipe has to be cut to profile. Pattern V shows the arrangement in a vertical pipeline. In all types of arrangements it should be ensured that the jumper joins the jacket pipe at minimum distance from the breakout flange in order to avoid cold / hot spots due to stagnancy. The jumper should also be provided with dismantling arrangement, either flange joints or unions, flange joint being preferred.

2.6 Spacers:

In order to keep the core pipe concentric with the jacket pipe, supports are provided at definite intervals. These are done by welding flat or bar to the core pipe called spacers. These spacers will be stitch welded to the core pipe. Flats are preferred as the restrictions in the jacket flow are minimum in this case. The arrangement shall be as shown in the sketch below:

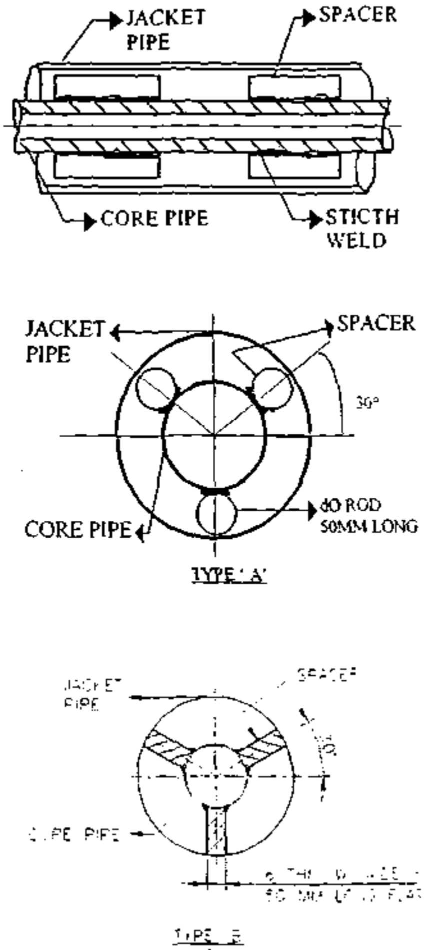


Fig. 2.4

NOTES: -

i) Material of spacers shall as that of the core pipe.

ii) Spacers near to pipe bend should be located at least 1000 mm away from the centerline of bend.

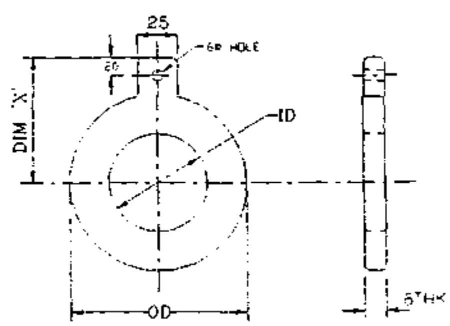
Spacer Details

Process pipe NB (mm)	Jacket Pipe NB (mm)	Dia of rod 'd' (mm)	Width of flat 'W' (mm)	Minimum span (mm)
15	40	8	-	1500
20	40	5	-	2000
25	50	8	-	2000
40	65	-	7	2500
50	80	-	8	3000
65	100	-	14	3000
80	100	-	6	3500
100	150	-	19	4000
150	200	-	16	5000

2.7 Jacket Barriers :

It is absolutely essential that the proper flow of the fluid in the jacket is maintained for proper heat transfer. Whenever there is a stagnancy or inadequate flow in the jacket, the hot spot or the cold spot gets formed which affects the process fluid flow in the core pipe. To avoid this, the fluid in the jacket may have to be directed properly. This is established by the provision of jacket barriers inside the jacket. These are mainly used where there are branch – offs. (Refer Fig. 2.5)

JACKET BARRIER



CORE PIPE SIZE	JKT PIPE SIZE	ID	OD	DIM 'X'
15	40	22.5	60	105
20	40	28.0	60	105
25	50	34.5	75	115
40	65	49.5	85	120
50	80	62.0	101	125
65	100	74.5	126	140
80	100	91.0	126	140
100	150	116.0	180	165
150	200	171.0	231	190
200	250	221.5	285	220

Fig. 2.5a

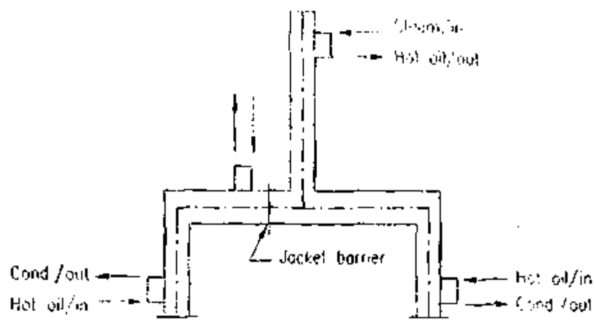


Fig. 2.5b

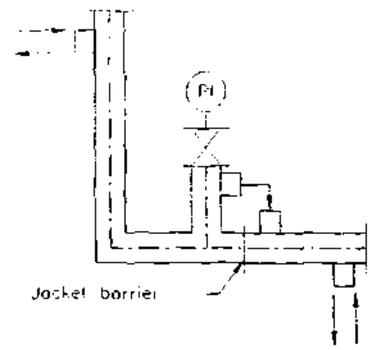


Fig. 2.5c

3.0 Fabrication:

Fabrication of the jacketed pipe shall be done with utmost care. The core pipe fabrication is more critical as the pipes when covered with jacket are not available for any visual inspection. The core pipe shall be assembled, welded and radiographed first. The jacket pipes and fittings shall be slipped over the core pipe in stages during assembly of core pipe. Sufficient gaps shall be left in the jacket pipe to inspect the core pipe during testing. After testing, inspection and acceptance of the core pipe, the gaps in jacket pipe shall be covered. There are two ways of covering these gaps.
One way is to fabricate a sleeve, which can slip on the jacket pipe, and these sleeves can be fillet welded over the jacket pipe. The other way is to split the jacket pipe into two longitudinal halves and insert it in to the gap left in the jacket pipe. This will need butt-welding the same with the jacket pipe (Fig. 2.6) and also the joints, in - situ.

The hub of the flanges are modified to provide seating for the jacket pipe. (Refer Fig 2.7)

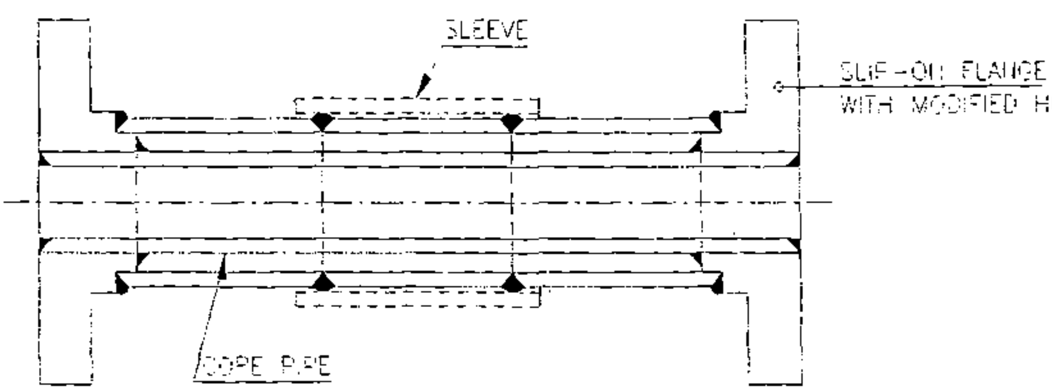
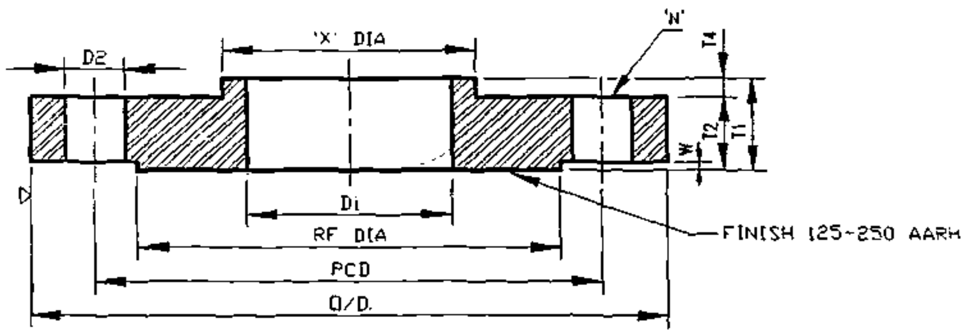


Fig.2.6

Reducing Flange with modified Hub



MOC — REFER PIPING SPECIFICATION.

Sr. No.	CORE PIPE	JACKET PIPE	FLANGE O/D.	PCD	UNBORED FLG. SIZE NB	'X' DIA FOR 106 GR.B SCH.40 PIPE	'X' DIA FOR IS:1239 HVY. CLASS	BORE D1	DIA BOLT HOLE D2	RF DIA	TOTAL THK. T1	T2	W	No. OF BOLTS 'N'	T4
1	10NB	25NB	108	79	25	24	25	18	14	51	18	14	1.6	4	4
2	15NB	40NB	127	98	40	39	37.5	22.5	14	73	22	16	1.6	4	4
3	20NB	40NB	127	98	40	39	37.5	28.0	14	73	22	18	1.6	4	4
4	25NB	50NB	152	121	50	50.5	48.5	34.5	18	92	25	19	1.6	4	6
5	40NB	65NB	178	140	65	60.5	64	49.5	18	105	29	22	1.6	4	7
6	50NB	80NB	190	152	80	76	76	62.0	18	127	32	24	1.6	4	8
7	65NB	100NB	229	190	100	100	100	74.5	18	157	33	24	1.6	8	9
8	80NB	100NB	229	190	100	100	100	91.0	18	157	33	24	1.6	8	9
9	100NB	150NB	279	241	150	152	151	116.0	22	216	35	25	1.6	8	10
10	150NB	200NB	343	298	200	201	201	171.0	22	270	41	29	1.6	8	12

Fig. 2.7a 150 # Rating Flanges

Sr. No.	CORE PIPE	JACKET PIPE	FLANGE O/D.	PCD	UNBORED FLG. SIZE NB	'X' DIA FOR 106 GR.B SCH 40 PIPE	'X' DIA FOR IS:1239 HVY. CLASS	BORE D1	DIA BOLT HOLE D2	RF DIA	TOTAL THK. T1	T2	W	No. OF BOLTS 'N'	T4
1	15NB	40NB	156	114	40	39	37.5	22.5	22	73	25	21	1.6	4	4
2	20NB	40NB	156	114	40	39	37.5	28.0	22	73	25	21	1.6	4	4
3	25NB	50NB	165	127	50	50.5	48.5	34.5	19	92	28	22	1.6	8	6
4.	40NB	65NB	190	149	65	60.5	64	49.5	22	105	32	25	1.6	8	7
5.	50NB	80NB	210	168	80	76	76	62.0	22	127	36	28	1.6	8	8
6.	65NB	100NB	254	200	100	100	100	74.5	22	157	41	32	1.6	8	9
7.	80NB	100NB	254	200	100	100	100	91.0	22	157	41	32	1.6	8	9
8.	100NB	150NB	318	270	150	152	151	116.0	22	216	47	37	1.6	12	10
9.	150NB	200NB	381	330	200	200.5	201	171.0	25	270	53	47	1.6	12	12

Fig. 2.7b 300 # Rating Flanges

4.0 Design of the Jacketed Piping System for Differential Expansion

When the materials of construction of the core and the jacket are different, it poses problem of differential expansion while in operation. The material, which has the higher coefficient of linear expansion, will try to pull the other one as both are rigidly fixed together at the flange joint. Consequently compressive stress will be developed in the material having higher coefficient of linear expansion and tensile stress gets developed in the material having lower coefficient of linear expansion. These stresses are to be calculated and compared with the allowable stresses at the operating temperature to ensure mechanical safety of the system. These stresses develop forces in the core and the jacket pipes and the magnitude of these forces remain the same since the system is in equilibrium. These forces are equated to calculate the distribution of the differential expansion. Based on the expansion, the strain and hence the stresses are calculated and compared with the allowable values. It is required that the jacket be trimmed at definite intervals so that the forces developed and hence the stresses are within the allowable limits. Based on the forces calculated as above, the jacket trimming distance is calculated.

5.0 Design Calculation For A Typical Jacket / Core Combination

A sample calculation of the jacketed piping system normally handled by design engineers is illustrated below.

5.1 *Data*

5.1.1 The pipe sizes under consideration are

a) Core -- 6" (150) NB.
b) Jacket – 8" (200) NB.

5.1.2 Materials of construction

a) Core – Austenitic stainless steel to ASTM A312 TP304L, seamless quality (for core pipes, always seamless quality is considered due to inaccessibility of the weld joint for inspection.)
b) Jacket – Carbon steel to ASTM A106 Gr. .B

5.1.3 Design Temperature

a) Core -- 700° F (≈ 375° C)
b) Jacket -- 750° F (≈ 400° C)

5.1.4 Design Pressure 1 kg/cm² = 14.28 psi

a) Core – 300 psig (21Kg/cm2)
b) Jacket – 400 psig (28 Kg/cm2)

5.1.5 Corrosion Allowance

a) Core – Nil
b) Jacket – 1/16" (1.6 mm)

5.1.6 Design Basis – ASME B31.3

5.2 Thickness Selection to without the Internal Pressures

{Refer ASME B31.3 clauses 304.1.1 and 304.1.2}

$$t_m = t + c$$

and
$$t = \frac{PD}{2(SE + PY)}$$

Where,

t_m = Minimum required thickness including mechanical, corrosion erosion allowances.

t = Pressure design thickness
c = Sum of mechanical, corrosion and erosion allowances
P = Internal design gauge pressure
D = Outside diameter of pipe
E = Quality factor from Table A – 1
S = Stress value from Table A – 1
Y = Coefficient from Table 304.1.1

a) Core Pipe:

P = 300 psig
D = 6.625″ (for 6″ NB)
S = 13500 psi (for SS 304L pipe at 700°F)
E = 1.0 (seamless quantity)
Y = 0.4

Hence,

$$t = \frac{300 \times 6.625}{2(13500 \times 1 + 300 \times 0.4)}$$

$= 0.073''$

$t_m = 0.073 + 0$

$= 0.073''$

Consider SCH 5S pipe as per ANSI / ASME B36.19

t = 0.109 ″

t (considering mill tolerance)
= 0.109 × 0.875
= 0.095 ″

Hence SCH 5S is adequate.

b) Jacket Pipe:

P = 400 psig
D = 8.625″ (for 8″ NB)
S = 13000 psi (for A 106 Gr. B pipe at 750° F)
E = 1.0 (seamless quality)
Y = 0.4

$$t = \frac{400 \times 8.625}{2(13000 \times 1 + 400 \times 0.4)}$$

$= 0.1311''$

$t_m = 0.1311 + 0.0625$

$= 0.1936''$

Consider SCH 20 pipe as per ANSI / ASME B36.10

t = 0.25 ″

t (considering mill tolerance)
= 0.25 × 0.875
= 0.2187 ″

Hence SCH 20 is adequate.

The above pipe selections were based on the internal design pressure of the core & jacket pipes.

5.3 Thickness Selection for the core pipe to withstand the external pressure

{Refer ASME Section VIII Division 1, UG – 28}

5.3.1. To check 6″NB SCH 5S pipe for an external pressure of 400 psig

$$\frac{L}{D_o} = 50$$

Thickness ‘t’ of SCH 5S pipe after mill tolerance = 0.095″ {Refer section 5.2 a) }

$$\frac{D_o}{t} = \frac{6.625}{0.095} = 69.7$$

Factor A = 0.000225
{Refer ASME Section II Part D}
Factor B = 2750
{Refer ASME Section II Part D Fig HA3}

Allowable Working Pressure,

$$Pa = \frac{4}{3} \times \frac{B}{D_o / t} = \frac{4}{3} \times \frac{2750}{69.7}$$

= 52.6 psig

Hence SCH 5S is not suitable.

5.3.2 To check 6″NB SCH 40S pipe for the external pressure of 400 psig.

$$\frac{L}{D_o} = 50$$

Thickness 't' of SCH 40S pipe after mill tolerance $= 0.28 \times 0.875$
$= 0.245''$

$$\frac{D_o}{t} = \frac{6.625}{0.245} = 27.0$$

Factor A $= 0.0015$
Factor B $= 4800$
Allowable Working Pressure,

$$Pa = \frac{4}{3} X \frac{4800}{27}$$

$= 237$ psig

Hence SCH 40S pipe is not suitable.

5.3.3 To check 6″NB SCH 80S pipe for the external pressure of 400 psig.

$$\frac{L}{D_o} = 50$$

Thickness 't' of SCH 80S pipe after mill tolerance $= 0.875 \times 0.432$
$= 0.378''$

$$\frac{D_o}{t} = \frac{6.625}{0.378} = 17.53$$

Factor A $= 0.0038$
Factor B $= 5500$

Allowable Working Pressure,

$$Pa = \frac{4}{3} \times \frac{5500}{17.53}$$

$= 418$ psig

Hence use SCH 80S pipe.

The selected combination shall be
Core of 6″ NB SCH 80S
(6.625 OD × 0.432 ″ nominal thk.)
Jacket of 8″ NB SCH 20
(8.625 OD × 0.25 ″ nominal thk.)

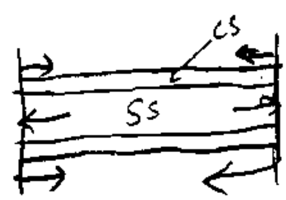
5.4 To check the Selected Combination of pipe thickness for stresses due to Differential Expansion.

The coefficient of linear expansion of carbon steel is less than that of the stainless steel. Hence the jacket pipe will restrict the expansion of the core pipe and the core pipe will try to pull the jacket on pipe. The differential expansion accordingly gets divided between the two. The proportion in which the differential expansion gets distributed can be calculated as below. The principle employed is that the differential expansion will develop stress and accordingly the force. There will be compressive force in the core pipe and tensile force in the jacket pipe. Since the system remains in equilibrium both the forces will have the same magnitude.

Strain due to differential expansion,

$$\varepsilon = \frac{\Delta l}{l} \qquad \text{.....................} 1$$

$$\text{Strain} = \frac{\text{Stress}}{\text{Modulus of elasticity}}$$



$$\varepsilon = \frac{f}{E} \qquad \ldots\ldots\ldots\ldots\ldots\ 2$$

$$\text{Stress, } f = \frac{\text{Force}}{\text{Area}} = \frac{P}{A} \qquad \ldots\ldots\ldots 3$$

$$\text{Hence, } E = \frac{f}{\epsilon} = \frac{P/A}{\Delta l / l}$$

$$\text{or} \quad P = E.\frac{\Delta l}{l}.A \qquad \ldots\ldots\ldots 4$$

Since, Force exerted by CS on SS and/or SS on CS is the same,

$$E_c \frac{\Delta l_c}{l} A_c = Es \frac{\Delta l_s}{l} A_s \ldots\ldots 5$$

Suffix 'c' stands for carbon steel and suffix 's' stands for stainless steel.
Hence,

$$E_c.\Delta l_c.A_c = Es.\Delta l_s.A_s \ldots\ldots\ldots\ldots 6$$

$$\text{or} \quad \frac{\Delta l_c}{\Delta l_s} = \frac{E_s A_s}{E_c A_c} \qquad \ldots\ldots\ldots\ldots 7$$

Hence,

$$\Delta l_c = \frac{E_s A_s}{E_c A_c} \Delta l_s \qquad \ldots\ldots\ldots\ldots 8$$

The Differential Expansion gets divided between carbon steel and stainless steel

i.e. $\Delta l = \Delta l_c + \Delta l_s$ 9

Substituting for Δl_c from above equation

$$\Delta l = \frac{E_s A_s}{E_c A_c}.\Delta l_s + \Delta l_s \ldots\ldots 10$$

$$= \Delta l_s \left(\frac{E_s A_s}{E_c A_c} + 1 \right) \qquad \ldots\ldots 11$$

$$\text{or} \quad \Delta l_s = \frac{\Delta l}{1 + \dfrac{E_s A_s}{E_c A_c}} \qquad \ldots 12$$

And $\Delta l_c = \Delta l - \Delta l_s$ .... derived from eqn. 9

Applying the above formulae in the example

Modulus of Elasticity of SS at 700° F
$E_s = 24.8 \times 10^6$ psi
(Refer ASME B31.3 Table C6)
Modulus of Elasticity of CS at 750° F
$E_c = 24.6 \times 10^6$ psi
(Refer ASME B31.3 Table C6)

Metal area (A_c) of 6" NB SCH 80S SS pipe
$= Pi/4\{6.625^2 - (6.625 - 2 \times 0.432)^2\}$ $\frac{\pi}{4}(d_o^2 - d_i^2)$
$= 8.405$ in^2
Metal area (A_c) of 8" NB SCH 20 CS pipe $\frac{\pi}{4}(d_o^2 - d_i^2)$
$= Pi/4\{8.625^2 - (8.625 - 2 \times 0.25)^2\}$
$= 6.578$ in^2
Expansion of CS pipe from 70° F to 750° F = 6.26"/100'
(Refer Appendix C of ASME B31.3)
Expansion of SS pipe from 70° F to 700° F = 7.50"/100'
(Refer Appendix C of ASME B31.3)

Differential Expansion,

Δl = 7.50 – 6.26 = 1.24"/100'

$\frac{\pi}{4} d^2$

Compression in ss pipe,

$$\Delta l_s = \frac{\Delta l}{1 + \dfrac{E_s A_s}{E_c A_c}} \quad \text{...(refer eqn. 12}$$

$$= \frac{1.24/100}{1 + \dfrac{24.8x10^6x8.405}{24.6x10^6x6.578}}$$

$= 0.542''/100'$

Tension in CS pipe,
$= 1.24 - 0.542 \quad = 0.698''/100'$

Strain in SS pipe,

$$\epsilon_s = \frac{0.542}{100 \times 12} = 0.000452$$

Strain in CS pipe,

$$\epsilon_c = \frac{0.698}{100 \times 12} = 0.000582$$

Tensile stress in CS pipe,
$= E_c \times \epsilon_c$
$= 24.6 \times 10^6 \times 0.000582$
$= 14317$ psi
$>$ Stress Allowable (S)

Compressive stress in SS pipe,
$= E_s \times \epsilon_s$
$= 0.000452 \times 24.8 \times 10^6$
$= 11210$ psi
$<$ Stress Allowable (S)

Hence SCH 20 Carbon steel pipe is not suitable for the service.
Increase the thickness of carbon steel pipe to SCH 40.

Nominal thickness of 8" NB SCH 40 pipe
$= 0.322''$
Metal area of 8" NB SCH 40 pipe
$= p/4\{8.625^2 - (8.625 - 2 \times 0.322)^2\}$
$= 8.399\ in^2$
Compression in ss pipe (refer eq.12)

$$\Delta l_s = \frac{1.24/100}{1 + \dfrac{24.8 \times 10^6 \times 8.405}{24.6 \times 10^6 \times 8.399}}$$

$= 0.617''/100'$

Tension in CS pipe
$= 1.24 - 0.617 = 0.623''/100'$

Strain in SS pipe,

$$\epsilon_S = \frac{0.617}{100 \times 12} = 0.000514$$

Strain in CS pipe,

$$\epsilon_c = \frac{0.623}{100 \times 12} = 0.000519$$

Tensile stress in CS pipe
$= 0.000519 \times 24.6 \times 10^6$
$= 12767$ psig
$<$ Stress Allowable (S)

Compressive stress in SS pipe
$= 0.000514 \times 24.8 \times 10^6$
$= 12747$ psig
$<$ Stress Allowable (S)

HENCE THE COMBINATION TO BE CONSIDERED FOR THE DUTY IS 6" (150mm) NB SCH 80S STAINLESS STEEL PIPE AND 8" (200mm) NB SCH 40 CARBON STEEL PIPES.

5.5 *To establish the maximum jacket trimming distance:*

It is necessary that the jacket is trimmed at definite intervals to ensure the stresses due to differential expansion do not exceed these values.

The stainless steel core pipe can be equated to a strut column with both ends fixed to establish the maximum distance allowed between two flanged joints.

Stress in ss pipe,

$f = 12747$ psi

Metal area of ss pipe,

$A = 8.405 \text{ in}^2$

Compressive force in the ss pipe,

$$P_c = f \times A$$
$$= 12747 \times 8.405$$
$$= 1{,}07{,}138 \text{ lbs}$$

Applying Euler's formula for column with both ends fixed

(Ref. Brownell & Young)

Compressive force,

$$P_c = \frac{4\pi^2 EI}{l^2} \quad \text{(eqn. 2.22 Table 2.1)}$$

Where,

I = Moment of Inertia
$= 40.49 \text{ in}^4$ $\frac{\pi}{64}(D_o^4-D_i^4)$

l = Distance between two supports in inches

Therefore,

$$1{,}07{,}138 = \frac{4\pi^2 \times 24.8 \times 10^6 \times 40.49}{l^2}$$

$$l = \sqrt{\frac{4\pi^2 \times 24.8 \times 10^6 \times 40.49}{1{,}07{,}138}}$$
$$= 608.29 \text{ inches}$$
$$= 50.7 \text{ ft } (15.45 \text{ m})$$

HENCE THE MAXIMUM JACKET TRIMMING DISTANCE SHALL BE 15450 MM.

6.0 STRESS ANALYSIS OF JACKETED PIPING

Unlike the stress analysis of normal piping systems where most of the checks are done by the software, the jacketed piping system, especially the discontinuous jacketing, needs some additional checks to ensure that the stresses developed are within the allowable limits.

6.1 While checking the stresses due to sustained loading and displacement strains as per clause 302.3.5 of ASME B31.3 or 102.3.2 of ASME B31.1, additional stresses developed due to the load at the junction of core and jacket i.e. P/A_c for core and P/A_j for jacket, should be added. The same philosophy is applicable while checking the limits of calculated stresses due to occasional loads as per clause 302.3.6 of ASME B31.3 or 102.3.3 of ASME B31.1 where P is the force at the junction of the core and the jacket and A_c and A_j are the area of the core and the jacket.

6.2 The weld strength between core and jacket also to be checked by considering an allowable load

P_{all}= Area of weld x 60% allowable stress.

Area of weld is obtained by multiplying the circumference of core pipe by the root of the weld i.e. π x d x (0.707 x weld size).

The force developed, available from the computer output, shall be less than the allowable value thus calculated.

6.3 The trimming length of the jacket shall be established ensuring that the buckling load calculated based on the Euler's formula is less than the load developed at the junction point of the jacket and the core as available from the computer output.

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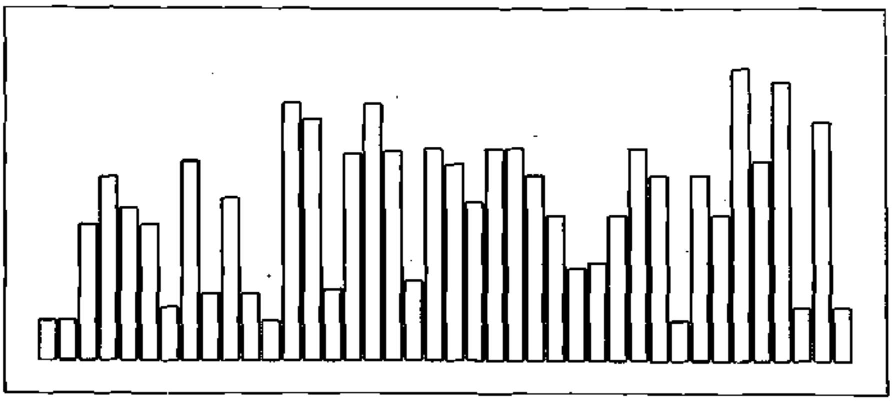
Certificate Course
on
PIPING ENGINEERING

June 12 - 25, 2006

DYNAMIC ANALYSIS

Prof. A. S. Moharir
IIT Bombay



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PIPING ENGINEERING CELL

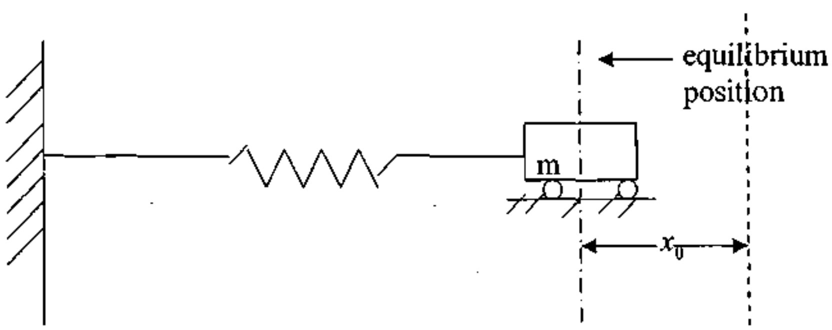
DYNAMIC ANALYSIS

Prof. A. S. Moharir

Piping systems in process plants are subjected to a variety of loads. It is customary to categorize these loads as sustained loads, thermal loads and occasional loads. Dynamic analysis pertains to the behaviour of piping systems under occasional loads. While the origin of these loads could be in process related or layout related or ambient related causes, these loads are typically oscillatory or impulsive in nature. These can be magnified due to resonance with the natural frequency of the piping systems or its specific segments. These can result into vibrations of large enough amplitude to cause severe bending moments and stress cyclicity, which eventually could lead to fatigue failure. Dynamic analysis is done to safeguard piping systems against such failures.

This articles attempts to prepare the reader with basic background to appreciate the causes of vibrations, their quantification, basic mathematical description of the dynamic phenomena, possible vibration arresters in real life and the code stipulation as regards the stresses that are allowable under combination of static, thermal and occasional loads.

Ideal Spring Mass System



Consider a massless spring attached at one end to a rigid support and at another end to a mass which can slide over a horizontal surface. Friction between the sliding mass and the surface is assumed to be zero.

While in this condition, the spring will be relaxed. The mass is not moving and hence the system has no kinetic energy as well. If some work is done on the system and it is pulled a distance x_0 in the horizontal plane, the work would be stored as energy of the system. The spring acquires potential energy ($= kx_0$, k being spring constant) which is equal to this work done. If the mass held stationary at x_0 is released now, it starts oscillating. The spring-mass system's energy is conserved during this oscillations. Only the potential energy of the spring and the kinetic energy of the mass interchange forms.

During oscillations, at any instant when the mass is at position x from the neutral position, one can write

Potential energy + Kinetic energy = Work done

Or

PE + KE = W = constant

Therefore,

$$\frac{d}{dt}(PE + KE) = 0$$

Considering the mass at position x at time t and its movement to a point $x + dx$ over time dt to attain a velocity from v at time t to $v + dv$ at time $t + dt$ and taking energy balance, one gets

$$mv + kx = m(v + dv) + k(x + dx)$$

or

$$mvdv + kxdx = 0$$

Integrating from $t = 0$ to t (i.e. $x = 0$ to x and $v = 0$ to v), one gets the sum of KE and PE at time t as

$$\frac{1}{2}mv^2 + \frac{1}{2}kx^2$$

Therefore, since $\frac{d}{dt}(KE + PE) = 0$,

$$m.v\frac{dv}{dt} + kx\frac{dx}{dt} = 0$$

As $v = \frac{dx}{dt}$ and $\frac{dv}{dt} = \frac{d^2x}{dt^2}$, we get

$$\left(m.\frac{d^2x}{dt^2} + kx\right)\frac{dx}{dt} = 0$$

This should be true for all t. As $\frac{dx}{dt}$ or velocity cannot be zero at every time (except when the mass is at its two extremes),

$$m\frac{d^2x}{dt^2} + kx = 0$$

or

$$\frac{d^2x}{dt^2} + \frac{k}{m}x = 0$$

The initial condition is

$$x = x_0 \text{ at } t = 0$$

$$\frac{dx}{dt} = 0 \text{ at } t = 0 \qquad \text{(i.e. at } x = x_0 \text{ or } -x_0)$$

The second order differential equation has a general solution

$$x = c_1 \cos\omega t + c_2 \sin\omega t$$

$$\frac{dx}{dt} = -c_1\omega \sin\omega t + c_2\omega\cos\omega t$$

Using the initial conditions,

$$c_1 = x_0 \text{ and } c_2 = 0$$

The solution describing the oscillation of the ideal mass-spring system is thus

$$x = x_0 \cos\omega t$$

and $$\frac{dx}{dt} = -wx_0 \sin\omega t\,.$$

Substituting the solution in the differential equation, one gets the circular frequency (ω)

$$\omega = \sqrt{k/m} \quad \text{radians/s}$$

The mass exhibits a harmonic motion with the following frequency and period.

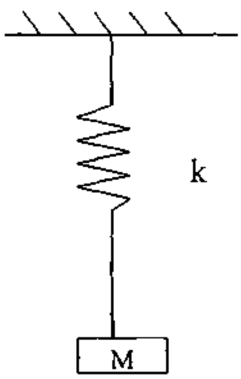
Frequency $f = \dfrac{1}{2\pi}\sqrt{k/m}$ cycles/s or Hz

Period $= 2\pi\sqrt{m/k}$ seconds.

The x vs t and $dx/dt = v$ vs t can be plotted. The ideal spring mass system thus undergoes undamped oscillations over infinite period, once it is disturbed/perturbed.

What happens if the mass is not at rest $(v = 0)$ when it is at x_0 at time $t = 0$ but has a velocity v_0?

What happens if the mass was hung to the spring vertically (see figure below) and then perturbed by a vertical push (up or down)?



What is the period of vibration and frequency of vibration?

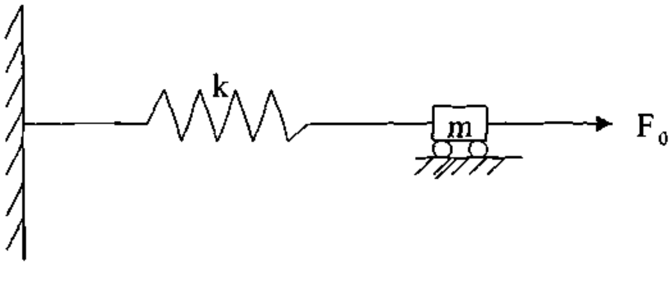
Why is the frequency of a spring-mass systems (k, m) called natural frequency?

A pipe has similarity with a spring mass system. Consider a pipe anchored between two supports. Under its own weigh, it may sag a little and attain equilibrium. Under this equilibrium, if a force upward, downward or sideways pulls it, the pipe wall strains and the work done in displacing the pipe gets stored as strain energy (which is a kind of potential energy). If the pipe is now released, it would oscillate at its natural frequency with the potential energy changing form to kinetic energy. If the pipe friction with ambient air or reaction at supports is negligible, these oscillations at pipe's natural frequency would continue for infinite duration. The frequency depends on the pipe layout, its mass (pipe content, insulation, piping elements, valves) and the properties of pipe and its material.

For the purpose of studying the oscillatory behaviour of a piping systems, perhaps it is possible to replace a pipe section by an equivalent spring-mass system with same frequency of vibrations. A spring-mass system is thus cardinal to understanding and modeling dynamic behaviour of in-plant piping.

External Force and Un-damped Oscillations

A piping system under industrial environment is always prone to vibration at its natural frequency. In addition to this, there could arise a sudden force/thrust on the system due to variety of reasons. Fluid force at bends, shift of end anchors (nozzles), relief, blowdown, etc. could cause such a force. The response of the pipe can be appreciated by considering the response of above spring-mass system subjected to a force (see figure).



Again, energy balance should give the equation governing this undamped spring mass system subject to an external steady force of (say) F_0.

Let the position of the center of the mass of the system be at x at time t and at $(x+dx)$ at time $t+dt$. Let the velocity be v and $(v+dv)$ at these two instants respectively. A modified balance taking into account that the mass has moved a distance x with force acting on it gives

$$mv + kx = m(v+dv) + k(x+dx) - F_0 dx$$

or

$$mvdv + kxdx = F_0 dx$$

Integrating as earlier from $t = 0$ to t (i.e. $x = 0$ to x and $v = 0$ to v), one gets

$$\frac{1}{2}mv^2 + \frac{1}{2}kx^2 = F_0 x$$

Differentiating with respect to t,

$$mv\frac{dv}{dt} + kx\frac{dx}{dt} = F_0\frac{dx}{dt}$$

As $v = \frac{dx}{dt}$ and cannot be zero all the time and with $\frac{dv}{dx} = \frac{d^2x}{dt^2}$, one gets

$$m\frac{d^2x}{dt^2} + kx = F_0$$

or

$$\frac{d^2x}{dt^2} + \frac{k}{m}x = \frac{k}{m}\frac{F_0}{k}$$

or

$$\frac{d^2x}{dt^2} + \omega^2 x = \omega^2 (F_0/k)$$

If the system was not oscillating, the force F_0 would have caused a deflection F_0/k. Let us call this static deflection (x_{stat}).

Therefore, the equation governing the undamped oscillations of a spring-mass system subjected to a steady force F_0 is

$$\frac{d^2x}{dt^2} + \omega^2 x = \omega^2 x_{stat}$$

s.t.

$$x = \frac{dx}{dt} = 0 \text{ at } t = 0.$$

The solution of the equation subject to these initial conditions is

$$x = x_{stat}(1 - \cos\omega t)$$

The systems thus oscillates between $x = 0$ (at $\omega t = 0$ or 2π) to $x = 2x_{stat}$ (at $\omega t = \pi$).

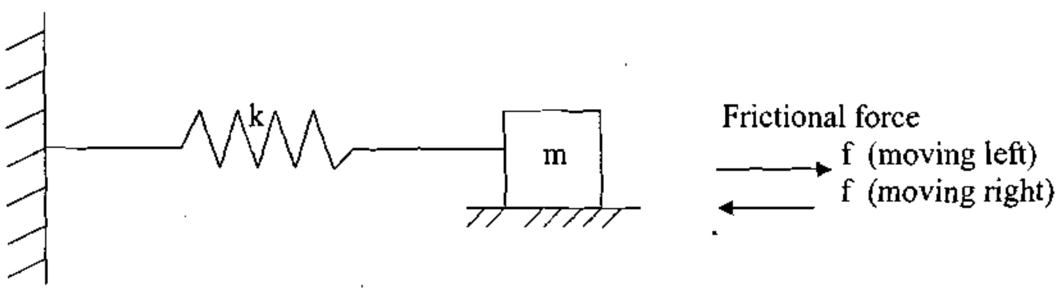
The ratio of maximum deflection during oscillations to static deflection is called Dynamic Amplification Factor (DAF) and is given in this case as

$$\text{DAF} = \frac{x_{max}}{x_{stat}} = 2.$$

Dynamic response in this case is twice the static response. This is an important observation.

Plot the deflection and velocity vs t. What are the amplitude, frequency and period of vibrations?

<u>Spring-Mass System with Friction</u>



Sliding of a mass in reality will encounter friction with the horizontal surface. Similarly, movement (oscillations) of a pipe would encounter friction with ambient air and end connections or supports. Energy will thus get dissipated with each cycle and one would expect the oscillating system to come to a halt after a finite number of cycles after initial perturbation.

This situation is similar to the previous case, but with a difference. The friction force acts on the mass in the direction opposing its motion. It thus acts from left to right as the mass swings from extreme right position to its extreme left position (first half cycle). It acts from right to left as the mass swings from leftmost to rightmost position (second half cycle)

Carry out the derivation and solution to arrive at the displacement vs time plot of these damped oscillations. Show that amplitude reduces by $(2f/k)$ in magnitude every

half period. Find the number of cycles for the oscillations to cease after they are set in motion by displacing the mass to the right by x_0 and releasing it. What is the frequency of oscillations?

Dynamics with Viscous Damping

The damping effect of frictional force is to confine the oscillation amplitude in an envelope defined by two converging straight lines. The damping force was of same magnitude although it changed sign at each turn of a half period. This kind of damping is also called Coulomb damping and is very useful in modelling dissipation of energy due to mechanical friction. Damping force is present in these cases as long as the mass moves and is of constant magnitude.

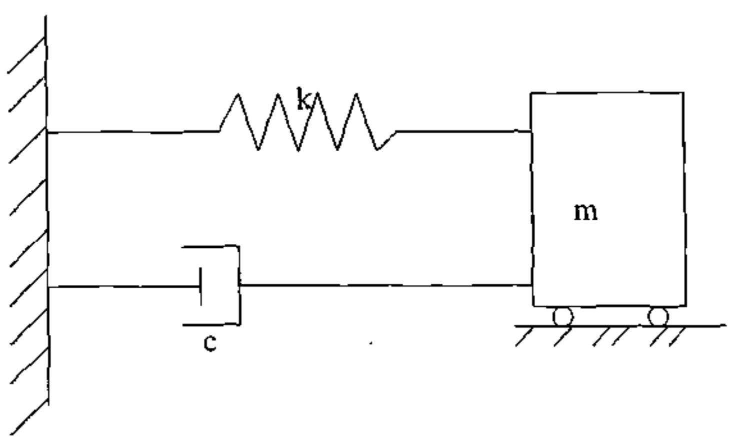
Another kind of damping force that is employed in practical situations is 'viscous' damping where the damping force itself is a function of velocity. Like Coulomb damping, the force acts in a direction opposite to the velocity, but its magnitude is also proportional to the magnitude of the velocity.

A good way to realize the difference between Coulomb damping force and the viscous damping force is as follows

Damping force $f_D = -f_C \dfrac{v}{|v|}$ Coulomb

$f_D = -cv$ Viscous

The governing equations and the solutions can be obtained for a spring-mass system with viscous damping.



Neglecting friction, the steps in derivation and results are presented here.

$$\frac{d}{dt}\left(\frac{1}{2}mv^2+\frac{1}{2}kx^2\right)=v\left(m\frac{dv}{dt}+kx\right)=f_Dv$$

$f_D=-cv$ (c is the damping coefficient of the dashpot)

$$m\frac{d^2x}{dt^2}+c\frac{dx}{dt}+kx=0$$

$$\omega=\sqrt{\frac{k}{m}} \quad , \quad \beta=\frac{c}{2m\omega}=\frac{c}{2\sqrt{km}}=\frac{c}{c_c}$$

c_c = critical damping coefficient.

$$x(t)=x_{\max}e^{-\beta\omega t}\sin(\omega_d t+\phi)$$

$$\omega_d=\sqrt{1-\beta^2}.\omega$$

$$x_{\max}=x_0\sqrt{\left[1+\left(\frac{v_0}{\omega_d x_0}+\frac{\beta\omega}{\omega_d}\right)^2\right]}$$

$$\phi=\tan^{-1}\left[\omega_d x_0/(v_0+\beta\omega x_0)\right]$$

The value of β is very important and decides the behaviour of the damped oscillator.

$\beta>1$ The system is overdamped. The solution is not periodic and involves only the exponential functions. Good examples of this are the devices used to stop doors from oscillating. Another example is the snubbers used at train terminals. Snubbers in piping systems are also in this category.

$\beta<1$ The system is underdamped.

$\beta=1$ The system is critically damped.

Draw the displacement vs time plots for all these cases.

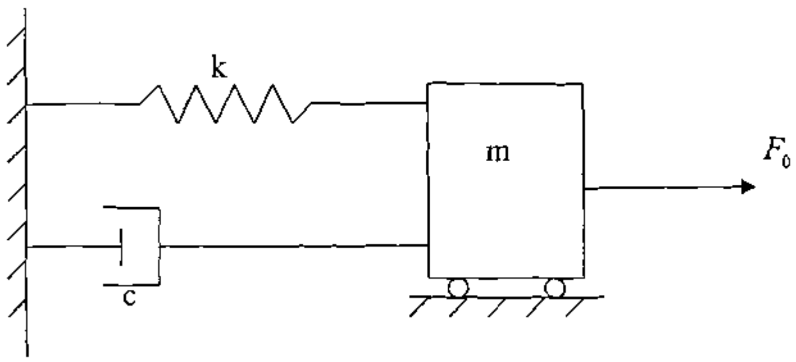
Can we write equations for cases where coulomb and viscous damping coexit.

Can we extend the analysis to cases where an external steady force also acts on the system?

What kind of damping can change the frequency of oscillations?

Which kinds of forces do not change frequency of oscillations?

Damped Oscillators and Sudden Force



$$\frac{d^2x}{dt^2}+2\beta\omega\frac{dx}{dt}+\omega^2x=\frac{F_0}{m}=\omega^2\frac{F_0}{k}$$

General solution

$$x(t)=e^{-\beta\omega t}\left(A\cos\omega_d t+B\sin\omega_d t\right)+\frac{F_0}{k}$$

Particular solution here is the static displacement and complimentary solution is the vibration solution with external force.

For $x=0$ at $t=0$ and $v=0$ at $t=0$,

$$x(t)=\frac{F_0}{k}\left[1-e^{-\beta\omega t}\left(\cos\omega_d t+\frac{\beta}{\sqrt{1-\beta^2}}\sin\omega_d t\right)\right]$$

The maximum displacement occurs at time t approximately equal to $\tau_d/2$. Its magnitude depends on β.

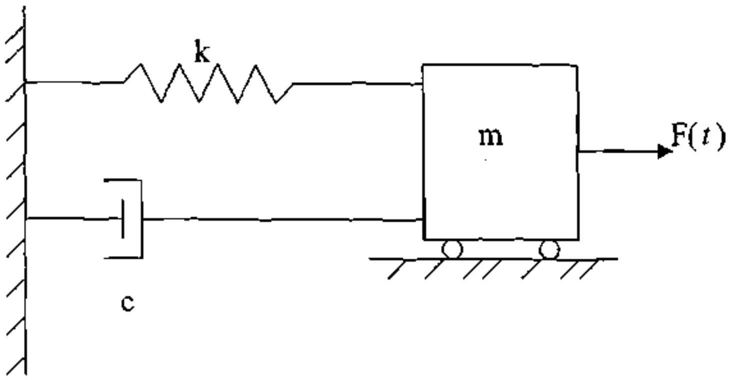
When $\beta=0$, $x_{max}=2F_0/k$
When $\beta=0.1$, $x_{max}=1.73F_0/k$

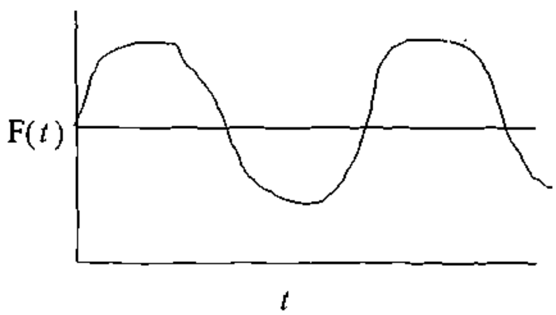
Damping values for most structures lie between 0.01 and 0.05, i.e. most structures are underdamped.

Observe that transients die out faster for higher natural frequency systems.

Damped Oscillator and Sinusoidal External Force

So far we have seen response of spring-mass systems to external force of a steady nature. Piping systems are also subject to external but periodic forces. These could be seismic, wind, due to attachment to vibrating equipment or flow induced. Often these are modeled as harmonic/sinusoidal forces.





Let the external force be

$$F(t) = F_0 \sin \Omega t$$

The equations and solutions are:

$$\frac{d^2x}{dt^2} + 2\beta\omega \frac{dx}{dt} + \omega^2 x = (F_0/m)\sin \Omega t$$

$$x(t) = x_{max} \sin(\Omega t - \phi)$$

$$x_{max} = \frac{(F_0 \; k)}{\sqrt{\left[\left\{1-\left(\frac{\Omega}{\omega}\right)^2\right\}^2 + (2\beta\,\Omega\;\omega)^2\right]}}$$

$$\phi = \tan^{-1}\left[2\beta\Omega\omega \Big/ \left\{1-\left[\frac{\Omega}{\omega}\right]^2\right\}\right]$$

The dynamic amplification factor is thus

$$\frac{x_{max}}{F_0 \; k} = \frac{1}{\sqrt{\left[\left\{1-\left(\frac{\Omega}{\omega}\right)^2\right\}^2 + (2\beta\,\Omega\;\omega)^2\right]}}$$

It is interesting to plot DAF vs the ratio of forcing function frequency and undamped natural frequency of the system. Create curves for various values of β (zero damping to heavy damping).

Consider two extremes, simplify equation of motion, solve it and interpret results.

1. $\Omega/\omega \to 0$
2. $\Omega/\omega \to \infty$

Also consider a case when frequency of the forcing function matches with the natural frequency of the system $(\Omega/\beta = 1)$. For this case, consider the effect of damping.

Consider what happens when there is no damping, critical damping.

What happens when β is in the range of most structures of interest.

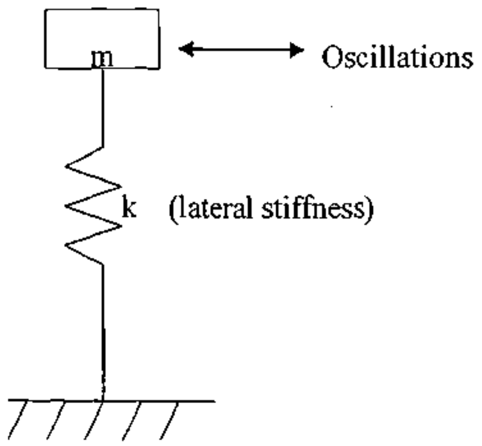
Appreciate the severity of the resonance phenomena.

If on the system, the external freq^n is equal to that of the natural freq^n, it may ~~allow~~ create large amplitude of freq^n it is called resonance

Snubber / viscous damper - It doesn't touch the piping system unless it is vibrating at resonance, it may simply absorb energy.

Another Spring-Mass System

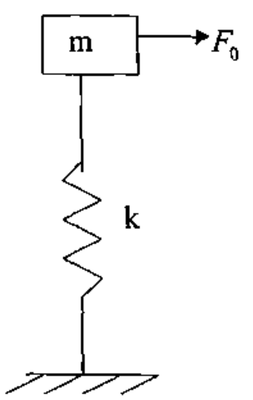
Consider a spring-mass system as shown.

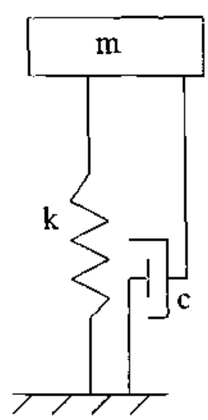


The spring is ideal and mass-less. The mass is perturbed in a lateral direction and oscillates. If a force F causes a displacement x at static equilibrium, F/x is called the lateral stiffness. Find the frequency and period of undamped oscillation.

Also analyse the following system with steady force F_0 acting on the mass in lateral direction.

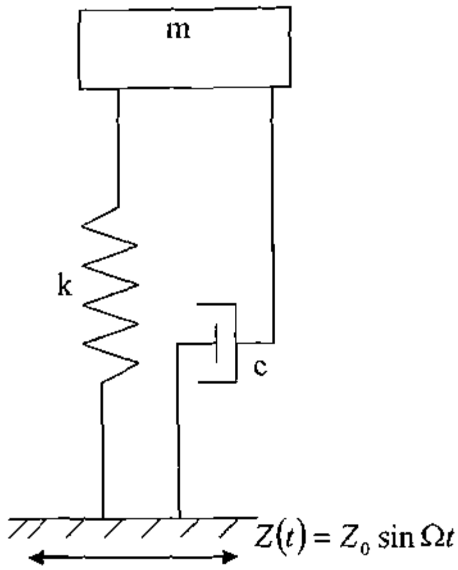
Also analyse a spring-mass system with viscous dampner.





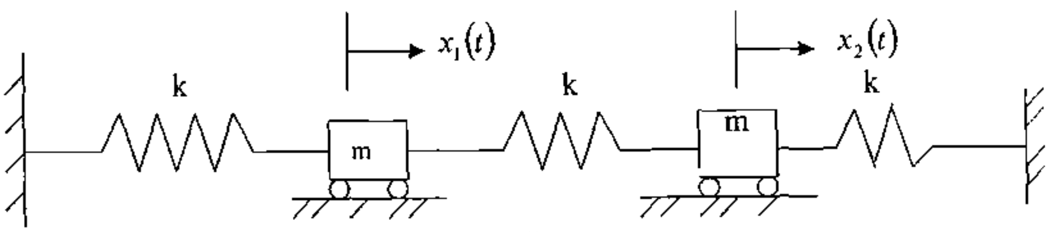
Stretch this to damped oscillator subject to steady force in lateral direction. Extend to sinusoidal force.

What happens if there is a sinusoidal base excitation imparted to a damped system?



More Complex Spring-Mass System

Consider a system with two masses and 3 springs. For simplification let the masses be same and springs identical.



Energy balance leads to the following condition

$$\frac{d}{dt}[P.E + K.E] = 0$$

$$\frac{d}{dt}\left[\frac{1}{2}mv_1^2 + \frac{1}{2}mv_2^2 + \frac{1}{2}kx_1^2 + \frac{1}{2}kx_2^2 + \frac{1}{2}k(x_2 - x_1)^2\right] = 0$$

$$v_1\left[m\frac{d^2x_1}{dt^2} + kx_1 - k(x_2 - x_1)\right] + v_2\left[m\frac{d^2x_2}{dt^2} + kx_2 + k(x_2 - x_1)\right] = 0$$

Each bracketed expression must vanish independently for this to be true. Thus,

$$\frac{d^2x_1}{dt^2} + \omega^2 x_1 - \omega^2 (x_2 - x_1) = 0$$

$$\frac{d^2x_2}{dt^2} + \omega^2 x_2 + \omega^2 (x_2 - x_1) = 0$$

or

$$\frac{d^2x_1}{dt^2} + 2\omega^2 x_1 - \omega^2 x_2 = 0$$

$$\frac{d^2x_2}{dt^2} - \omega^2 x_1 + 2\omega^2 x_2 = 0$$

Visual inspection of the governing equations indicate that two modes of oscillations of the system can satisfy the system of equations.

Mode 1 : $x_1(t) = x_2(t)$
Mode 2 : $x_1(t) = -x_2(t)$

Let us interpret these modes. In mode 1, we hold both the masses and move them a distance x_0 to the right (or left) and leave them. Symmetry of the situation tells us that the masses would oscillate in phase with same frequency and amplitude.

In mode 2, we hold both the masses, move them towards each other by a distance x_0 each and then leave them to oscillate. Symmetry again tells us that the two masses would oscillate with same amplitude and frequency, but out of phase.

Let in mode 1, the two masses oscillate with a frequency ω_1. The expected solution is then

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 1 \\ 1 \end{bmatrix} x_0 \cos \omega_1 t$$

(i.e. $x_1 = x_2 = x_0 \cos \omega_1 t$)

Substituting this in any of the two equations of motion, we get

$$\omega_1^2 = \frac{k}{m} \Rightarrow \omega_1 = \sqrt{k/m}$$

So the first natural frequency of vibration is

Q1 Frequency amplitude is 2 to 3 Hz — provide snubbers
above freq >33 Hz its safe

$$f_1 = \frac{\omega_1}{2\pi} = \frac{1}{2\pi}\sqrt{k/m}$$

Similarly, in the second mode, let the frequency be ω_2. Therefore

$$\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 1 \\ -1 \end{bmatrix} x_0 \cos\omega_2 t$$

Substituting in the equation of motion, one gets

$$\omega_2^{\ 2} = \frac{3k}{m} \Rightarrow \omega_2 = \sqrt{\frac{3k}{m}}$$

The second natural frequency of vibration is

$$f_2 = \frac{\omega_2}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{3k}{m}}$$

The system thus appears much stiffer while vibrating in mode 2 than in mode 1. Can one offer an explanation for this?

Visualization of the modes for this spring-mass system was quite easy. For more complex system, this may not be. We therefore look for some mathematics behind identification and definition of the modes.

The two equations of motion can be written in the matrix form as

$$\begin{bmatrix} \dfrac{d^2x_1}{dt^2} \\ \dfrac{d^2x_2}{dt^2} \end{bmatrix} + \frac{k}{m}\begin{bmatrix} 2 & -1 \\ -1 & 2 \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

The solution vectors for mode 1 and were $\begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 1 \\ 1 \end{bmatrix} x_0 \cos\omega_1 t$ and $\begin{bmatrix} 1 \\ -1 \end{bmatrix} x_0 \cos\omega_2 t$ respectively. The vectors $\begin{bmatrix} 1 \\ 1 \end{bmatrix}$ and $\begin{bmatrix} 1 \\ -1 \end{bmatrix}$ are eigen vectors of the stiffness matrix $\begin{bmatrix} 2 & -1 \\ -1 & 2 \end{bmatrix}$. Each eigen vector thus indeed defines a mode.

In a more complex system, the stiffness matrix will have larger dimensions and larger number of eigen vectors and modes of vibration. Frequency of each mode can be found out by using the solution vector defined by eigen vector for that mode. Natural frequencies of vibration can then be defined, arranged in ascending order of stiffness.

Dynamic Analysis of Piping Systems

We have now all the ingredients that go in the dynamic analysis of a complex piping system.

The piping system is defined for the stress analysis s/w to carry out dead weight analysis and static flexibility analysis. For dynamic analysis, this is then converted to an equivalent spring-mass network. The mass and spring system with appropriate values attached to mass and spring stiffness simulate the vibration characteristics of a piping section between any two modes. Coulomb damping, viscous damping can be added wherever necessary. The system under analysis is thus transformed to a 3-D spring-mass network with spring straining possible in axial/lateral direction. Modes of vibration are then identified through eigen vector approach. Behaviour of the system under steady force, sinusoidal force or base excitation can then be studied. These external forces/displacements could be for variety of reasons such as seismicity, wind load, water hammer, relief, blowdown, machine vibrations, etc. Appropriate modes would resonate and cause displacements. Maximum displacement of a section is used to calculate bending moments and stresses at nodes. These must be within allowable limits stipulated by codes.

If stress anywhere crosses allowable limits, frequency of vibration is altered to remove or reduce resonance possibilities. These measures could be by additional supports, snubbers, etc.

A system safe from dead weight, thermal flexibility and occasional load considerations is then passed for construction.